Power and The Engineer

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THE ability to reveal our thoughts and emotions to others by word of mouth is given to most of us. It is comparatively easy to communicate with one another in this way because the voice can be so modified and inflected, accentuation and tone can be so varied as to impart the most subtile shades of meaning. In addition to these vocal variables there are the stage settings—so to speak—facial expression and gesticulation. And yet, in spite of all these, how often are our remarks misunderstood! Who has not had the sad experience of unwittingly offending or hurting the feelings of some good friend by a well meant joke or bit of persiflage.

Think then, what a difficult thing it must be to make a written statement perfectly clear. Few who have not been put to the test realize how hard a job it is to write *exactly* what is intended so that the reader will *get* what is meant. Simple thoughts or ideas, of course, do not offer much difficulty. But, let the subject matter be extensive enough to require several consecutive sentences, and the trouble will begin.

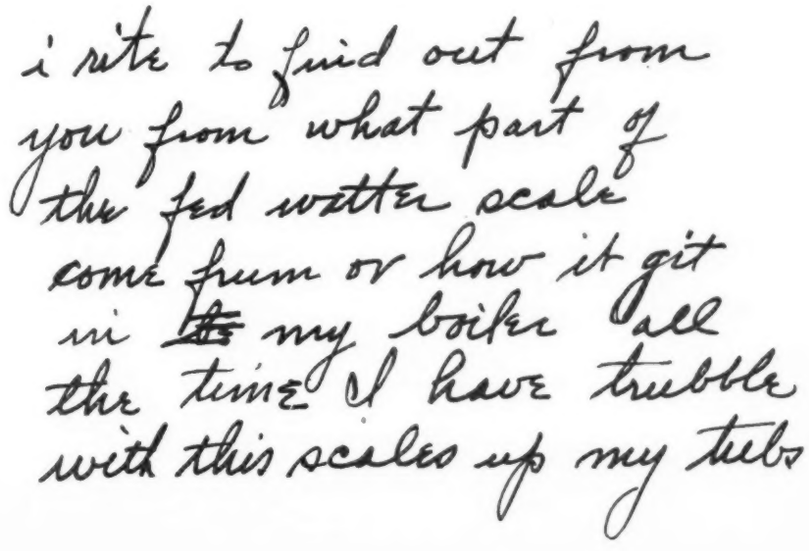
These very paragraphs will be misconstrued by hundreds. It probably is no exaggeration to say that each one who reads them will find a meaning at least slightly different from that found by anyone else.

Are all agreed, then, that it is hard to write correctly?

* * * * *

There is more credit and benefit in doing *with a fair degree of* success something which is difficult, and improving the performance each time, than there is in doing something which is easy and requires little effort. Therefore, when you get an idea—an idea which you want to give to others or an idea that you want help—put it into writing, the best writing you can muster, and send it to us. The practice will be good, for it is practice that makes perfect. If you do not come off with first "honors," what of it; you have had the practice.

The writing shown herewith is fictitious but is a fair sample of letters occasionally received at this office. The writer at least conveys his meaning and shows that he is interested in his work. Therefore we are always glad to receive such a letter.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When a man gets busy with the pencil, no matter how poor a speller or grammarian he may be, hope is not lacking. There *must* come a time when he will be better at spelling and at a good many other things.

The first thing is to get busy; the next, to keep busy.

Compound Engine Proportions

By Alwin Hofmann

By means of the diagrams any one of the factors—initial pressure, back pressure, cylinder ratio, cutoff and mean effective pressure referred to the low-pressure cylinder—may be determined if the others are given. This refers to both compound condensing and noncondensing engines.

For a number of years the writer has been using the accompanying diagrams in connection with calculations relating to compound engines, and as they have been found of considerable help they are given herewith with the thought that they may prove useful to engineers and others who are engaged in this kind of work.

It is now common practice to proportion compound engines by figuring first the low-pressure cylinder, as if all the work were done in that cylinder. After this has been done, the high-pressure cylinder is calculated in accordance with the desired cylinder ratio or the kind of work which the engine is called upon to perform, as will be explained later.

On the left-hand vertical scale of the diagrams, of which one is for condensing and the other for noncondensing engines, the cylinder ratio is given, and on the right-hand vertical scale the mean effective pressure referred to the low-pressure cylinder; that is, the mean effective pressure which would be obtained in that cylinder if all the work were done in it. The curves plotted across the diagrams represent cutoffs in the high-pressure cylinder, and the straight lines represent different gage pressures at the throttle.

The diagrams can be used for a number of different purposes, which can best be explained by citing some practical examples of their use. First, consider the case of an engine where it is desired to determine the mean effective pressure that can be developed with a known steam pressure. Suppose this is a condensing engine carrying 26 inches vacuum, 150 pounds gage steam pressure at the throttle with a cylinder ratio of 4 to 1 and cutoff at one-quarter stroke. Looking on the left-hand scale of the diagram, Fig. 1, opposite the cylinder ratio of 4 to 1, follow the horizontal line to its intersection with the one-quarter cutoff curve. From this point proceed vertically as far as the steam-pressure line of 150 pounds, then horizontally to the right-hand scale where will be read a mean effective pressure of 28 pounds referred to the low-pressure cylinder. The pressure lines are marked for gage, not absolute pressures.

The chart for noncondensing engines is used in the same manner. The mean effective pressures represent the actual figures which one may expect to obtain from an engine with Corliss or similar type of valve gear running at a piston speed of 600 to 800 feet per minute, allowance having been made for cylinder clearance, loss of pressure through ports, etc.

The vacuum is taken as 26 inches at the exhaust flange of the engine, but for each inch increase or decrease in vacuum, 0.38 of a pound can be added to or subtracted from the mean effective pressure obtained.

The engines are assumed to have neither reheating receivers nor steam-cylinder jackets. Where a reheating receiver is used, 4 per cent. can safely be added to the mean effective pressures obtained. Where both reheater and steam jackets are used, 10 per cent. should be added to the mean effective pressures.

For piston speeds of less than 600 to 800 feet per minute, the mean effective pressure will increase slightly. For high-speed engines or similar types the mean effective pressure may be reduced by about 5 per cent.

The noncondensing diagram is figured on the basis of the low-pressure cylinder exhausting freely into the atmosphere at an altitude of approximately sea level. Where an engine operates against a back pressure, the mean effective pressures should be increased about 0.85 pound for each pound of back pressure. At altitudes above sea level, 0.4 pound per 1000 feet altitude should be added.

The numbers given on the diagrams along the steam-pressure lines are the absolute pressures which will be obtained at the end of expansion in the low-pressure cylinder for any mean effective pressure. Attention is called to the fact that this terminal pressure is not dependent upon the cylinder ratio nor the cutoff, but is determined solely by the steam pressure and ratio of expansion, or the mean effective pressure referred to the low-pressure cylinder. The terminal pressures are especially useful in proportioning noncondensing engines, as will be explained later.

The second use of the diagrams is that of determining the cylinder sizes of a compound engine, and this can also be best explained by giving a few examples. In proportioning the cylinders, the first thing to do is to figure the size of the low-pressure cylinder—the cylinder ratio is an after consideration—the important thing being to figure the low-pressure cylinder so as to have the proper mean effective pressure referred to that cylinder in accordance with the results desired. Too much stress cannot be laid upon this point, because many engineers are inclined to overlook this.

The cylinder ratio alone is no indication of economy of an engine. A high cylinder ratio may or may not mean an economical engine, and under certain conditions a high ratio may even be a detriment and reduce the economy rather than help it. For instance, consider a condensing engine designed so as to require a comparatively high mean effective pressure referred to the low-pressure cylinder. If a high-pressure cylinder which will give a high cylinder ratio is used here, it will not help the economy because the high-pressure cylinder will then operate with a late cutoff and therefore the port friction will be excessive; in other words, the high-pressure cylinder is too small and the ports therefore not large enough to properly handle the amount of steam needed to do the work.

No definite rule can be laid down giving the proper mean effective pressure upon which to figure the low-pressure cylinder, as this depends upon the results desired. For instance, in the case of a condensing engine, if a highly economical engine is wanted a high ratio of expansion should be used or what amounts to the same thing, the mean effective pressure referred to the low-pressure cylinder should be low. This does not mean that the lower the mean effective pressure the higher the economy, because a point is soon reached where increasing the ratio of expansion or reducing the mean effective pressure adds nothing to the economy on account of the increased amount of cylinder condensation encountered when using a high ratio of expansion.

A high ratio of expansion also means a large low-pressure steam cylinder, and therefore high first cost. On this account, it is the engineer's business to so choose the mean effective pressure as to make the engine commercial, or to design it in accordance with the degree of economy desired. Consider a condensing engine operating at 150 pounds steam pressure and 26 inches vacuum. If a highly economical engine is desired without regard to first cost, it would be good practice to use a mean effective pressure of about 25 pounds, or a ratio of expansion of, approximately, 19. Increasing the ratio of expansion above this would not improve the economy appreciably, therefore it would be bad practice to go beyond this. After deciding upon the mean effective pressure and knowing the horsepower required, the stroke and the speed, it is a simple matter to determine the diameter of the low-pressure cylinder from the following formula:

$$I.h.p. = \frac{D^2 \times N \times S \times P}{252{,}000}$$

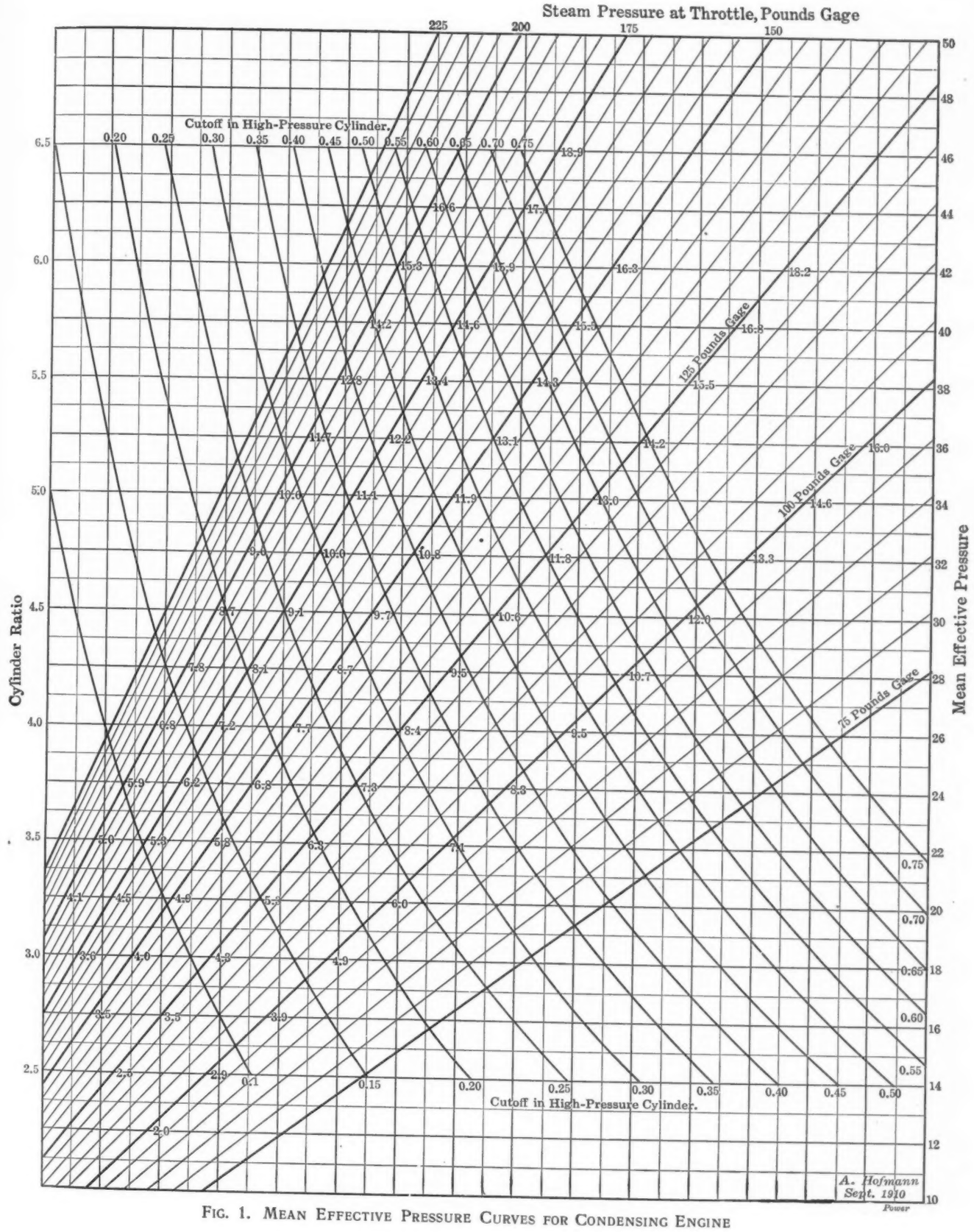


FIG. 1. MEAN EFFECTIVE PRESSURE CURVES FOR CONDENSING ENGINE

where,

$I.h.p.$ = Indicated horsepower;
D = Diameter of the low-pressure cylinder in inches;
N = Revolutions per minute;
S = Stroke in inches;
P = Mean effective pressure in pounds, referred to low-pressure cylinder.

In determining the size of the high-pressure cylinder, the cutoff in that cylinder is first decided upon. This should not occur too early in the stroke, as in that case the cylinder condensation will be excessive; and it should not occur too late in the stroke or the pressure losses through the ports will be too great. In the case in hand, where economy is the first consideration, a cutoff of between 25 and 30 per cent. will give good results. Referring to the diagram, Fig. 1,

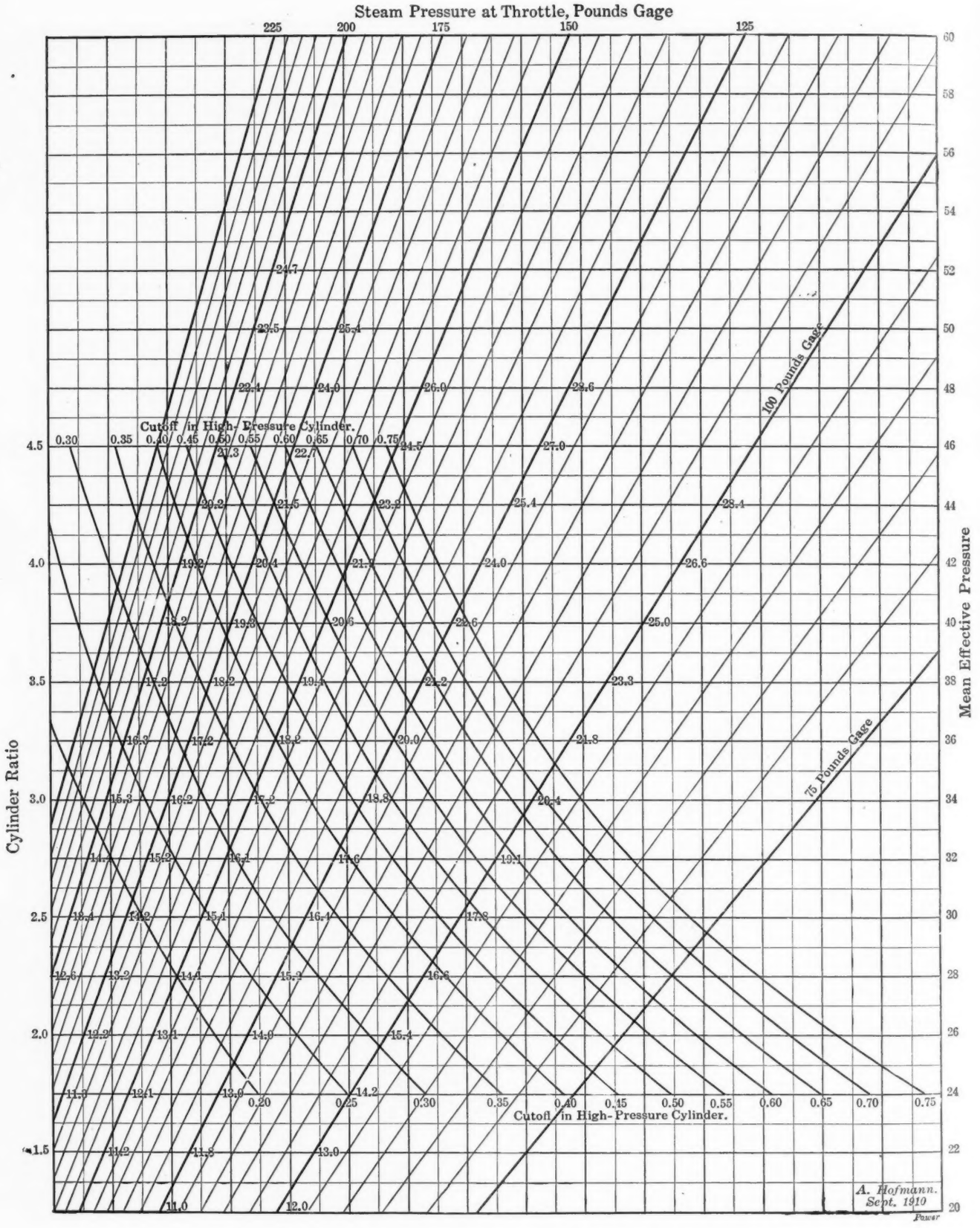


FIG. 2. MEAN EFFECTIVE PRESSURE CURVES FOR NONCONDENSING ENGINES

it will be found that with 25 pounds mean effective pressure, 150 pounds steam pressure and, say, 27½ per cent. cutoff, the cylinder ratio will be 5 to 1. Having determined the cylinder ratio, it is a simple matter to figure the diameter of the high-pressure cylinder, the size of the low-pressure cylinder having already been found.

In the case of the foregoing example, it has been assumed that the engine is to operate with a comparatively constant load. On the other hand, if the load is variable a higher mean effective pressure should be used in order that when the engine is operating under a light load the condensation losses will not be excessive, also with a variable load a com-

paratively low cylinder ratio should be used so as to give an early cutoff at normal load, thus giving the engine a large overload capacity.

If, instead of a highly economical engine a commercial engine is desired, that is, one in which economy may be sacrificed in favor of price, the low-pressure cylinder is proportioned by using a higher mean effective pressure. This requires a smaller low-pressure cylinder. In the case of the example cited, a mean effective pressure of 28, or even 30 pounds, could be used, and a cylinder ratio of 4 to 1, which would give a good engine but still not one of maximum economy.

In figuring the cylinders of a noncondensing engine, the operation proceeds along somewhat different lines. The main object here is to have the steam expand down close to atmospheric pressure in the low-pressure cylinder, for when this condition exists maximum economy is obtained. Therefore, in determining the proper mean effective pressure in the case of a compound noncondensing engine, it is necessary only to find the terminal pressure given along the steam-pressure lines in the diagram, and then choose the mean effective pressure so as to give a terminal pressure of about 15 pounds or a little higher, in which case the steam will expand down practically to atmospheric pressure; that is, there will neither be a loop in the low-pressure indicator card nor excessive drop, neither of which is desirable in a noncondensing engine. The size of the high-pressure cylinder can be determined the same as in the case of condensing engines, except that with a noncondensing engine a later cutoff must be used in the high-pressure cylinder; otherwise that cylinder will be too large and most of the work will be done in the high-pressure cylinder instead of being divided equally between the two cylinders.

Another use of the diagrams which is evident, but which attention is specially called to here, is the following: Assume an existing engine with a cylinder ratio of 3.5 to 1, operating noncondensing, the indicated horsepower to be developed being such as to require a mean effective pressure of 28 pounds, referred to the low-pressure cylinder. It is desired to know if this engine will operate at 125 pounds steam pressure, provided it has a cutoff gear which will permit only a cutoff at 40 per cent. of the stroke and no later. Referring to the diagram, Fig. 2, it will be found that under these conditions the cutoff in the high-pressure cylinder will be just a little short of 40 per cent. and therefore the engine will operate at 125 pounds steam pressure.

Milwaukee Garbage Destructor

This destructor plant is one of the latest to be put in operation in this country. The garbage is incinerated without the aid of fuel other than that found in the refuse matter. The heat from the destructor is used to generate steam in four 200-horsepower boilers supplying the engine equipment of the plant. Excess steam is allowed to go to waste.

Disposing of the garbage, ashes and refuse matter collected from cities is a problem that has cost thousands of dollars with little or no commercial returns to show for the outlay. Various methods have been employed in disposing of these materials, but have proved unsatisfactory, due to the high cost of removal or the failure of the destructor apparatus to give satisfactory results. Milwaukee has probably had the most varied experience in the disposal of its garbage of any city in this country. Almost every known method of garbage disposal has been tried at various times during the last twenty years. Prior to 1882 different methods were used in different wards, the people either burying the garbage, feeding it to poultry, hogs and cattle or dumping it on vacant lots.

These methods were abolished in 1882, when a municipal collection system was installed and an experimental destructor was built which was unsuccessful. The garbage was then dumped in adjoining towns and later in the lake. Then another destructor was built and an attempt was made to use oil and coal to burn the garbage. This plant was operated for six years, when another type of reduction plant was built and operated until closed by the health commissioner in 1890. For two years the garbage of the city was dumped into the lake and then discontinued as it was found to contaminate the water supply.

Then for five years the garbage was removed by water transportation, at the end of which period a system of municipal collection and burying farm was again installed. In 1901 another crematory was built to use coal for fuel, which was operated on and off up to May 12 of this year.

It has been found in England, Germany, Belgium and France that all domestic refuse when mixed will burn without additional fuel. Due to this experience and influenced by a report on the disposal of garbage by Dr. Rudolph Hering, the city of Milwaukee advertised for bids under specifications which besides the usual guarantee of capacity, freedom from dust, smoke or other nuisance, required guarantees of the amount of labor and the quantity of useful steam to be delivered by the plant. A contract was awarded The Destructor Company of New York City for the construction of a Heenan refuse destructor of 300 tons daily capacity, and the plant has been operated by the Board of Public Works since June 17, 1910.

The plant consists of four independent units, each of 75 tons daily capacity, arranged in the four corners of a rectangular two-story and basement building.

Fig. 1 is an exterior view of the plant from the lake side. Fig. 3 is a cross-sectional view through the building showing the general arrangement of the furnaces. Fig. 2 is a view of the clinkering floor where the operation of the plant is carried on. The superintendent's office, electrical machinery and recording instruments are located on this floor. The upper story is used entirely as a

FIG. 1. VIEW OF PLANT FROM THE LAKE SIDE

feeding floor and for storage of refuse, see Fig. 4, which is hoisted into the building and distributed into the various storage bins by a pair of overhead electrically operated traveling cranes. The basement is used for a clinker railway and doors for cleaning the dust from the furnace.

The refuse is collected in two kinds

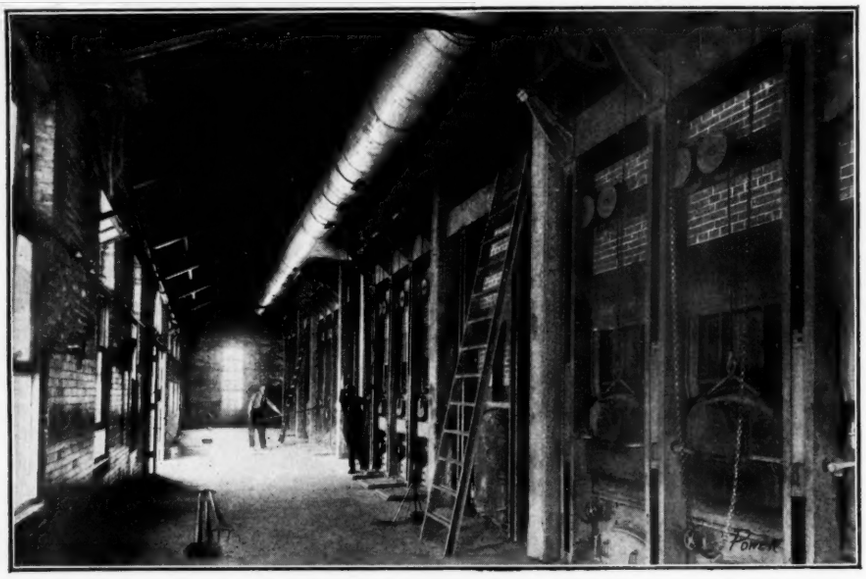


FIG. 2. VIEW OF CLINKERING FLOOR

of carts. One type has a removable steel body of about 1½ cubic yards capacity and is used chiefly for collecting garbage. The body is hoisted by a crane and the contents discharged into bins on the feeding floor. The other type of cart is bottom dumping, of about 2½ cubic yards capacity. These carts are used chiefly for the collection of rubbish and ashes and dump into boxes resting in pits at the ground level; the boxes are picked up by the cranes and their contents transferred to the storage bins. From the storage bins, the men who feed the furnaces rake or shovel the material into the mouths of rectangular steel chutes termed "containers." These containers are located directly over the top-feed openings into the furnaces, the sliding doors to the furnace openings also serving to keep the bottoms of the containers closed.

The man feeding from above keeps the containers charged with mixed material at all times and the doors admitting the contents into the furnaces are controlled by the fireman from the floor below. The material when dropped into the furnace by the operation of the sliding doors, falls on a drying hearth where a large proportion of the moisture is carried away with the hot gases. The fireman rakes the dried material forward on the grates as required, for which purpose small stoking doors are provided.

After a sufficient amount of material has been burned on each grate to form a thick body of clinker, the large counterweighted clinkering doors in the front are opened and the clinker, after being broken up by bars, is drawn out of the furnace and dropped through holes in the floor into steel dumping cars on tracks below, whence it is pushed outside the building and allowed to cool

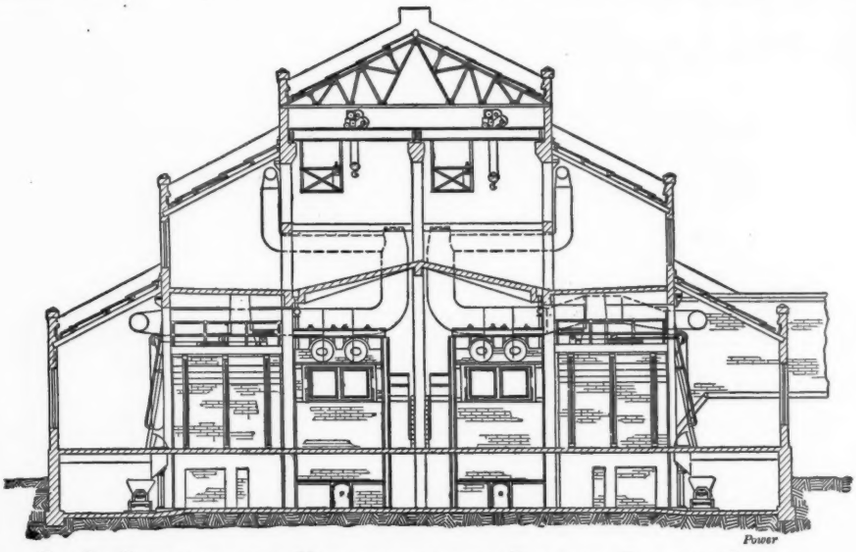


FIG. 3. CROSS-SECTIONAL VIEW THROUGH THE BUILDING, SHOWING GENERAL ARRANGEMENT

FIG. 4. FEEDING FLOOR

on the ground. Fig. 5 is a half-plan view of the furnace and clinkering floor.

Each furnace consists of six separate grates located three on each side of a central combustion chamber through which the products of combustion of all six grates pass and are thoroughly mixed. The velocity through this combustion chamber is so slow that the combustion is entirely completed and dust is allowed to settle. After leaving the combustion chamber, the gases from each unit pass over the heating surface of a 200-horsepower water-tube boiler, and then through an air heater or regenerator on their way to the chimney.

The combustion is entirely controlled by forced draft, all the air being drawn from ventilating ducts in the building by an engine-driven fan, and raised to a temperature of over 300 degrees in the air heater. Fig. 6 shows the path of the gases and air supply from the ven-

tilating ducts. The proportions are such that the amount of air required for combustion when the plant is operating at full capacity is seven times the cubic

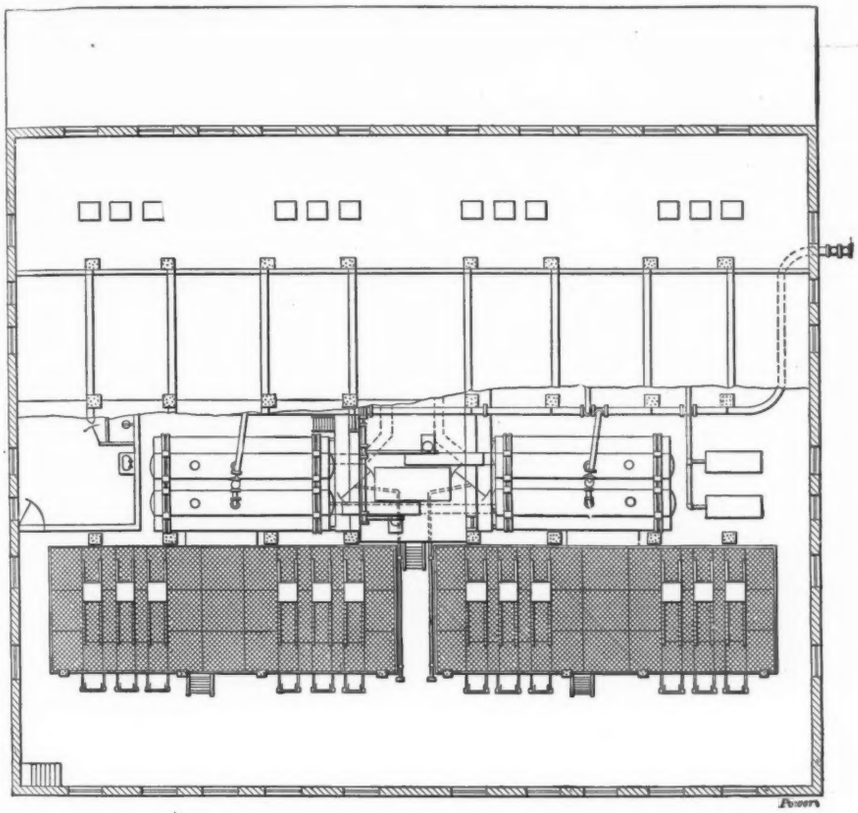


FIG. 5. PLAN OF HALF OF FURNACE AND CLINKERING FLOOR

contents of the building, which insures the renewal of all the air in the building about every eight minutes, and prevents the escape of any gases from the building except through the chimney. Separate ashpits are provided under each grate with controlling valves, so that the rate of combustion and the amount of air supplied may be varied under the individual grates at will. Each unit is fitted with its own fan and engine with independent flues.

At present garbage is taken from the entire city and refuse and ashes from but seven out of 24 wards. This makes the percentage of garbage to be disposed of in the destructor exceedingly high on account of lack of combustible matter. The average annual mixture collected has the composition given in the table.

RESULTS OF TESTS MADE ON PLANT.

Garbage, percentage	41
Ashes, percentage	41
Rubbish, percentage	5
Manure, percentage	13
Dates of Tests: May 26, 27, 30, 31, June 1, 1910.	
Quantity of refuse consumed in 37 hours, tons	126.81
Contract requirement, in 40 hours, tons	125.00
Excess over guaranteed capacity, per cent.	9.66
Rate of burning per sq.ft. area, lb.	64.00
Steam evaporated from and at 212 degrees Fahrenheit per pound refuse destroyed, lb.	1.34
Guaranteed evaporation per pound refuse destroyed, lb.	1.1
Excess over guaranteed evaporation: lb.	0.24
percentage	21.8
Temperature of combustion chamber, average, degrees Fahrenheit	1664
minimum, degrees Fahrenheit	1267
Temperature of air supply, degrees Fahrenheit	351
COST OF OPERATION:	
Labor per unit of 75 tons rated capacity per 24 hr:	
One feeder, 37 hr. at 25c	$ 9.25
Three firemen, 37 hr. at 25c	27.75
¼ engineer's time, 37 hr., 37½c.	3.47
Total cost of labor	$40.47
Cost of labor per ton	0.319
Steam required to operate fan engine and feed pump, per hr. at 130 pounds gage pressure, lb.	694
Equivalent from and at 212 degrees per hr., lb.	843
Steam consumption per ton, lb.	19
Value of steam consumed per ton of refuse incinerated at contract price of 4 cents per 100 lb.	$0.088
Labor	0.319
Total gross cost per ton	$0.407
Revenue for steam per ton of refuse incinerated at contract price of 4 cents per 100 lb. for 2680 lb. of steam	$1.072
Net profit per ton	$0.665

The tests show that the cost of steam obtained from incinerating one ton of refuse at a contract price of four cents per 100 pounds for 2680 pounds of steam is $1.072, but as a matter of fact the steam made by the boilers in the plant is not used for any commercial purpose other than operating the engine driving the necessary machinery throughout the plant. The excess steam is allowed to go to waste through a reducing valve into a sump filled with rocks to deaden the sound.

It costs the city of Milwaukee $25,000 a year to operate two flushing pumping plants and one sewerage pumping station. It is stated on good authority that the steam allowed to go to waste from the destructor boiler into the sump would be sufficient to operate these three pumping stations, providing the necessary equipment were procured to utilize this power, which would not exceed an estimated amount of from $75,000 to $100,000.

A similar destructor plant to be located on the opposite side of the city has been advocated. This would not only double the possible power output from the disposal of garbage, but would decrease to an appreciable extent the present cost of cartage.

At a recent meeting of the Institution of Municipal and County Engineers in Dublin, says *The Surveyor*, a communication of considerable value by M. Sellars, the town surveyor of Dundalk, on the scraping of water mains was submitted for consideration. In cases where the action is corrosive, of course, the material of the pipe must be adversely affected; while in cases of mere incrustation or a surface deposit there may at the same time be a damaging influence at work on the pipe. As Mr. Sellars points out, the principal cause of reduction of delivery from a pipe which has become incrustated or corroded is not to be found in the reduced bore, but in the roughness of surface which is a necessary accompaniment of a pipe so affected. This roughness produces sudden eddies and cross currents which, even apart from the great amount of head which is lost in friction, tends to an unduly large diminution of velocity. This, then, is the evil, and the remedy points toward efficient pipe scraping, which will thoroughly remove this incrustation.

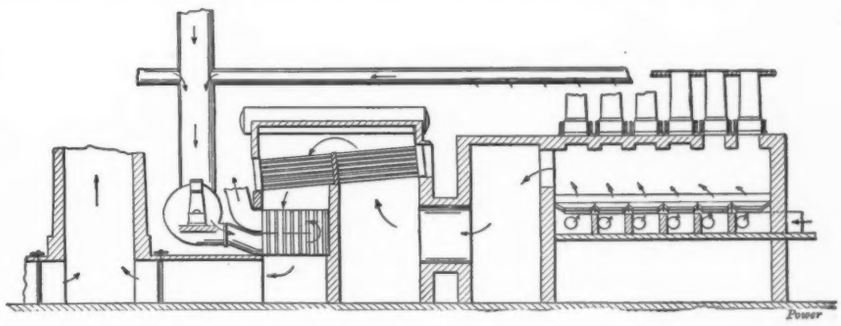


FIG. 6. SHOWING PATH OF GASES AND AIR SUPPLY FROM VENTILATING DUCTS

Factors Affecting Air Pump Capacity

By R. M. Neilson

Although it is common practice to determine the capacity of air pumps by arbitrary rules, such methods should always be checked by calculations based upon the behavior of a mixture of air and vapor under the conditions existing in the condenser.

Air pumps for handling the exhaust from steam engines are often designed by rule-of-thumb methods. There is some excuse for this practice because it is often difficult to ascertain the probable volume of air and vapor which the pump will be required to handle per minute; but with large units the cost of the air pump is such a significant factor that it is worth while to ascertain the minimum size of pump which will suffice. As the function of the air pump is to withdraw the air from the condenser, it would seem that for any vacuum the capacity of the air pump—the effective piston displacement per minute—should be proportional to the air which enters the condenser per minute. The air pump, however, necessarily withdraws steam or water vapor along with the air, and this fact must be taken into account.

Air in a condenser is always associated with steam or water. It is well known that there is a definite temperature corresponding to any pressure of saturated steam. The steam cannot be reduced in temperature without reducing the pressure, and cannot be increased in temperature—unless by being superheated—without being accompanied by a rise in pressure. However, if the steam is in intimate association with water it cannot be superheated.

A fact of equal importance to steam engineers, but less widely known, is that for any mixture of air and steam in definite proportions there is a definite temperature for every pressure. The proportion of the mixture, that is, the ratio by weight of air and steam, cannot be altered by condensing some of the steam unless either the temperature or the pressure of the mixture is altered. To explain the reason of this would involve a dissertation upon the laws of partial pressures of mixed gases, which is somewhat beyond the scope of this article.

It is important to note that the higher the temperature for any given pressure, the greater is the proportion of steam to air, the limit being reached when the temperature is that of the boiling point of water at the given pressure, at which point the ratio of steam to air is infinite. Lower temperatures than that corresponding to any given pressure represent a reduced ratio of steam to air; hence it is desirable to reduce the temperature of the mixture which is to be discharged by the air pump so as to reduce the size of the latter and the power required to drive it. Manufacturers and users of condensers in general are aware of the advantage of reducing the temperature of the mixture; and condensers, both of the injection and the surface type, are now usually arranged on the contraflow principle, so that the mixture of air and vapor, just before leaving the condenser, is cooled by the coldest entering water. In addition to this the air is often passed through a special cooler placed between the condenser and the air pump.

It is often stated that the reason for seeking to thus reduce the temperature of the mixture is to reduce the volume of the air, because the volume of a gas is proportional to its absolute temperature. The real reason, however, is that a reduction in temperature reduces the amount of steam or water vapor associated with the air. As steam, this occupies a large volume, but when condensed into water, its volume is almost negligible. The mere reduction in volume of dry air by reducing the temperature a few degrees is of little account. If dry air at a pressure of 2 inches of mercury is reduced in temperature from 95 to 85 degrees Fahrenheit, the diminution in volume is only 1.8 per cent; but, if air saturated with water vapor at this pressure is reduced in temperature from 95 to 85 degrees Fahrenheit, the volume will be diminished 56 per cent.

The accompanying tables show the volumes of air-steam mixtures such as are handled by air pumps, at vacuums ranging from 24 to 29 inches, referred to a 30-inch barometer. For example, Table 1 deals with a 24-inch vacuum; this means that the vacuum is 24 inches when the barometer stands at 30 inches. Moreover, the table is applicable whenever the vacuum is 6 inches less than the barometer; hence, it would apply when the barometer is at 29 inches and a mercury gage indicates that the vacuum at the exit end of the condenser is 23 inches.

The air contained in water, used for boiler feed or for condensing purposes prior to entering the pump, varies in amount up to about 4 per cent. by volume. Water is actually incapable of retaining air in solution to the extent of 4 per cent., but the mixture of gases found in water differs more or less from atmospheric air, although usually spoken of as such. Moreover, water often temporarily contains more air than it can permanently hold in solution.

A boiler-feed pump or condensing-water pump having air-tight glands and drawing its supply from below a quiescent water level can add nothing to the air already contained in the water.

In the first column of each table is given the percentage by volume of air in water, and the second column expresses this relation by weight. Column *A*, Table I, gives the volume of the air at a temperature of 121 degrees Fahrenheit if removed from the water and perfectly free from water vapor at a vacuum of 24 inches. The figures in this column are in themselves of no practical use in connection with a condensing plant, as the air which has to be dealt with by the pump is never free from water vapor. This column, however, is given to show to what extent the volume of air is augmented by the water vapor.

In column 3, Table I, the conditions as regards temperature and vacuum are the same as in column *A*, but the air is saturated with water vapor. This is the condition of the air when handled by the air pump. By comparing the figures in columns 3 and *A*, the large increase in volume due to the water vapor will be apparent. The amount of water vapor is, however, relatively small at this temperature. In column 5, where the temperature is 10 degrees higher, the volumes are nearly doubled. This increase in volume is due not so much to the increased temperature of the air — this would account for only about 2 per cent. increase in volume — as to the much greater proportion of water vapor which is associated with the air at this temperature. In column 7 the temperature is high and the volume of air and vapor greater. The various columns in the succeeding tables express similar conditions to those in Table I. Figures corresponding to those in column *A*, Table I, are not given in the succeeding tables because they have no direct bearing on the present discussion.

In a jet condenser the air contained in the feed water is of relatively small importance, as a much larger quantity enters the condenser along with the condensing water. In deciding upon the capacity of an air pump for a jet condenser, the average percentage of air in the injection water should be ascertained, if possible, and if there is much variation in the percentage, the maximum percentage should be ascertained. Then from the weight of condensing water to be pumped per minute the re-

TABLE 1. 24-INCH VACUUM.

1	2	A	3	4	5	6	7	8
Air in Water.		Volume of Same Air at 121 Degrees F. and 24 Inches Vacuum, Dry. Cubic Feet per 100 Cubic Feet of Water.	Volume of Same Air Saturated with Water Vapor at 121 Degrees F., and at 24 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 131 Degrees F., and at 24 Inches Vacuum		Volume of Same Air Saturated with Water Vapor at 135 Degrees F., and at 24 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water		Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	2.795	6.88	0.001103	12.18	0.00195	19.5	0.003125
1.0	1.226	5.590	13.76	0.002206	24.36	0.00390	39.0	0.006250
1.5	1.84	8.385	20.64	0.003309	36.54	0.00585	58.5	0.009375
2.0	2.45	11.180	27.52	0.004412	48.72	0.00780	78.0	0.012500
2.5	3.07	13.975	34.40	0.005515	60.90	0.00975	97.5	0.015625
3.0	3.68	16.770	41.28	0.006618	73.08	0.01170	117.0	0.018750
3.5	4.29	19.565	48.16	0.007721	85.26	0.01365	136.5	0.021875
4.0	4.90	22.360	55.04	0.008824	97.44	0.01560	156.0	0.025000

TABLE 2. 26-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 106 Degrees F. and at 26 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 116 Degrees F., and at 26 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 120 Degrees F., and at 26 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	9.8	0.00158	17.4	0.00279	29.4	0.00472
1.0	1.226	19.6	0.00315	34.8	0.00558	59.0	0.00945
1.5	1.84	29.5	0.00473	52.4	0.00838	88.5	0.01419
2.0	2.45	39.3	0.00630	69.5	0.01115	117.9	0.01888
2.5	3.07	49.2	0.00789	87.2	0.01399	147.8	0.02365
3.0	3.68	59.0	0.00945	104.8	0.01678	176.9	0.02835
3.5	4.29	68.7	0.01101	121.8	0.01950	206.0	0.03302
4.0	4.90	78.5	0.01260	139.2	0.02230	235.5	0.03775

TABLE 3. 27-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 95 Degrees F. and at 27 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 105 Degrees F., and at 27 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 109 Degrees F., and at 27 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	11.98	0.00192	21.0	0.00337	32.3	0.00518
1.0	1.226	23.95	0.00384	42.0	0.00674	64.6	0.01036
1.5	1.84	35.55	0.00570	63.0	0.01012	96.9	0.01556
2.0	2.45	47.90	0.00768	84.0	0.01348	129.2	0.02070
2.5	3.07	60.00	0.00962	105.0	0.01685	161.5	0.02595
3.0	3.68	72.00	0.01152	126.0	0.02022	193.8	0.03110
3.5	4.29	83.70	0.01342	147.0	0.02360	226.1	0.03620
4.0	4.90	95.80	0.01535	168.0	0.02695	258.4	0.04145

TABLE 4. 27.5-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 89 Degrees F., and at 27.5 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 99 Degrees F., and at 27.5 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 103 Degrees F., and at 27.5 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	13.91	0.00223	24.96	0.00400	40	0.00641
1.0	1.226	27.82	0.00447	49.92	0.00800	80	0.01282
1.5	1.84	41.73	0.00670	74.88	0.01200	120	0.01923
2.0	2.45	55.64	0.00894	99.84	0.01600	160	0.02564
2.5	3.07	69.55	0.01117	124.80	0.02000	200	0.03205
3.0	3.68	83.46	0.01341	149.76	0.02400	240	0.03846
3.5	4.29	97.37	0.01564	174.72	0.02800	280	0.04487
4.0	4.90	111.28	0.01788	199.68	0.03200	320	0.05128

TABLE 5. 28-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 81.5 Degrees F., and at 28 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 81.5 Degrees F., and at 28 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 95.5 Degrees F., and at 28 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	17.22	0.00276	30.6	0.00490	49	0.00785
1.0	1.226	34.44	0.00552	61.2	0.00980	98	0.01570
1.5	1.84	51.66	0.00828	91.8	0.01470	147	0.02355
2.0	2.45	68.88	0.01104	122.4	0.01960	196	0.03140
2.5	3.07	86.10	0.01380	153.0	0.02450	245	0.03925
3.0	3.68	103.32	0.01656	183.6	0.02940	294	0.04710
3.5	4.29	120.54	0.01932	214.2	0.03430	343	0.05495
4.0	4.90	137.76	0.02208	244.8	0.03920	392	0.06280

TABLE 6. 28.5-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 72 Degrees F., and at 28.5 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 82 Degrees F., and at 28.5 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 86 Degrees F., and at 28.5 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	21.15	0.00339	36.95	0.00592	60.5	0.00968
1.0	1.226	42.30	0.00678	73.90	0.01184	121.0	0.01936
1.5	1.84	63.45	0.01017	110.85	0.01776	181.5	0.02904
2.0	2.45	84.60	0.01356	147.80	0.02368	242.0	0.03872
2.5	3.07	105.75	0.01695	184.75	0.02960	302.5	0.04840
3.0	3.68	126.90	0.02034	221.70	0.03552	363.0	0.05808
3.5	4.29	148.05	0.02373	258.65	0.04144	423.5	0.06776
4.0	4.90	169.20	0.02712	295.60	0.04736	484.0	0.07744

TABLE 7. 28.8-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 65 Degrees F., and at 28.8 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 75 Degrees F., and at 28.8 Inches Vacuum.		Volume of Same Air Saturated with Vapor at 79 Degrees F., and at 28.8 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	26.6	0.00426	45	0.0072	71.8	0.0115
1.0	1.226	53.2	0.00852	90	0.0144	143.6	0.0230
1.5	1.84	79.8	0.01278	135	0.0216	215.4	0.0345
2.0	2.45	106.4	0.01704	180	0.0288	287.2	0.0460
2.5	3.07	133.0	0.02130	225	0.0360	359.0	0.0575
3.0	3.68	159.6	0.02556	270	0.0432	430.8	0.0690
3.5	4.29	186.2	0.02982	315	0.0504	502.6	0.0805
4.0	4.90	212.8	0.03408	360	0.0576	574.4	0.0920

TABLE 8. 29-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 59.5 Degrees F., and at 29 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 69.5 Degrees F., and at 29 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 73.5 Degrees F., and at 29 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	31.25	0.00501	57	0.00913	84.5	0.01355
1.0	1.226	62.50	0.01002	114	0.01826	169.0	0.02710
1.5	1.84	93.75	0.01503	171	0.02739	253.5	0.04065
2.0	2.45	125.00	0.02004	228	0.03652	338.0	0.05420
2.5	3.07	156.25	0.02505	285	0.04565	422.5	0.06775
3.0	3.68	187.50	0.03006	342	0.05478	507.0	0.08130
3.5	4.29	218.75	0.03507	399	0.06391	591.5	0.09485
4.0	4.90	250.00	0.04008	456	0.07304	676.0	0.10840

quired capacity of the air pump can be calculated. An allowance must be made for air entering the condenser with the steam. The amount of such air depends chiefly upon the tightness of the glands and joints. With a low vacuum and cold injection water, containing less than 2 per cent. air, the weight of air which enters the condenser with the injection water is relatively small and may be less in amount than that which enters with the steam, even if there is no excessive leakage; but, as a rule, and especially with high vacuum, the air which enters with the injection water constitutes the greater proportion of the total air; in the case of a steam-turbine condensing plant, this may constitute over 75 per cent. of the total air.

Where jet condensers are used the air and water are discharged by separate pumps and a certain amount of the air is discharged with the water. In the case of barometric jet condensers where the water is discharged through a tail pipe without the use of a pump, a certain proportion of the air is carried away by the water. The amount of work which the air pump is thus relieved of depends upon the design of the condenser.

The displacement per minute of the air-pump piston does not represent the effective capacity of the pump, which is the product of the piston displacement and the volumetric efficiency, the latter varying greatly.

In a surface condensing plant the greater part of the air handled by the air pump enters the system as leakage, consequently the weight of air per pound of steam varies greatly. If only the air contained in the feed water had to be dealt with by the air pump, the latter could be of much less capacity than is usually necessary. Therefore, columns 1 and 2 of the tables are of little use in connection with air pumps for surface condensers which usually discharge the water of condensation as well as the air and vapor. The tables are useful, however, in connection with a surface condensing plant chiefly to show how the volumes of air and vapor are affected by the vacuum and temperature in the condenser.

Although it is practically impossible to determine with great precision the necessary dimensions of an air pump, a rational treatment of the subject should precede the actual design; in this way many serious blunders may be avoided. It is no exaggeration to say that in some cases air pumps are made 100 per cent., and occasionally 200 per cent., too large, while in other cases they are much too small. A pump intended to maintain a certain vacuum, but of too small a capacity, will suck all the air as fast as it enters the condenser, but it will do so only when the vacuum in the condenser has fallen so much below the intended vacuum that the volume of the air and vapor is reduced to an amount corresponding to the effective capacity of the pump. By comparing the various tables with each other it will be apparent that the volume decreases very rapidly with the vacuum.

The Confession of an Engineer

By R. O. Warren

Most engineers have failed to make good at some time because they did not use their knowledge in a practical way. In this story the engineer never thought of keeping records and computing the coal consumption per horse-power-hour. The manager did.

After a man has spent twenty-seven years in an engine room it would be strange if he could not remember almost countless instances where mistakes had been made, either through ignorance or lack of judgment, and lost opportunities for making good because of lack of interest and enterprise.

It was only a few days ago, while sitting in my engine room that I began to recall some of the mistakes I have made during my engineering career. The longer I thought of these things the more I became impressed with the fact that the numerous errors I had committed and the greatest mistakes I had made had been due to failure in putting the knowledge I possessed to practical use to the advantage of both my employer and myself.

Such musings added greater weight to the conviction I already had that a man can know a whole lot about engineering, but if he fails to use that knowledge he is worth no more to his employer than a man who knows less, but uses what knowledge he does possess in a practical way.

I had had about three years' experience as an engineer at the time of which I speak, and, although I knew considerable about power-plant operation, a habit had been formed of letting well enough alone so long as things were moving along apparently in a satisfactory manner. No greater mistake is ever made by an engineer than when he gets the idea that conditions cannot be improved. No steam plant has ever reached such a point of perfection that it cannot be made better and more efficient. If every engineer would only fix this fact firmly in mind early in his career, he would be a much better engineer and of more value to his employer.

I did not see the matter in this light during my early engineering training, but got the lesson after passing through many humiliating experiences which, of course, lowered my standing in the eyes of my employer. About the first intimation I had that the manager was looking after his interests in the engine room occurred one day after I had been with the Holden Manufacturing Company about eight months. The firm had followed for years the practice of purchasing the coal supply from a local coal dealer. My past experiences have removed all doubt that most coal dealers fail to deliver the grade of coal ordered and seldom if ever duplicate an order with the same grade of coal, although it is sold under the same name and at the same price.

Evidently the new manager had had some experience with engineers who knew things and had acted upon their advice, for upon the day in question he dropped into the engine room and asked in a casual way how the coal was holding out. I answered that there was enough to last about four weeks, but that there was no need of getting anxious because if the new coal did not arrive on time, Miner & Hall, the coal dealers, would supply all we wanted and that it did not make much difference one way or the other as they furnished all the coal we used anyway.

Upon receiving this information, Wood, for that was the new manager's name, sat down and, after lighting a cigar, asked in an "I-am-not-interested" way how much coal I used in getting a horsepower out of the engine.

Now I knew how to figure the horsepower of an engine for I had worked out hundreds of such problems. Not only that, but I owned an indicator outfit and could take a card in rather an expert manner. So familiar was I with the operation and the figuring that the formula

$$\frac{PLAN}{33{,}000} = I.H.P.$$

was one of the easiest that could have been proposed.

Being energetic in some ways, I had expounded this very formula and covered the blackboard with figures down at the association time and time again, and yet the new manager had me stumped at the first question he had asked. For I did not know how many pounds of coal I was burning per horsepower developed by the engine. Can anyone imagine a more humiliating position for an engineer to occupy?

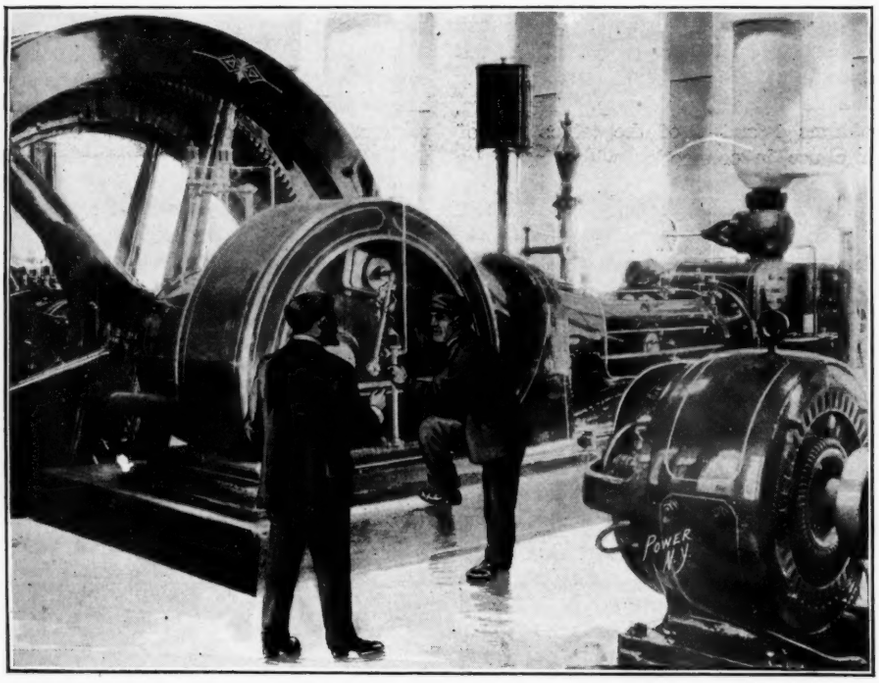
I was not man enough to own up to

my ignorance and tried to give the impression I knew by answering, "Oh, about four pounds." I had heard someone say that was a good average.

"Four pounds!" exclaimed Wood, "you don't mean that you are burning four pounds of coal for each horsepower developed by your engine, running condensing at that? Why, that is excessive."

"Ah, it might be a little less," I replied, feeling mighty uncomfortable, for dim recollections of having read of two pounds of coal for a condensing engine began to flit through my brain.

"But you are not sure," said he, "and now, honest, Warren, did you ever take enough diagrams from your engine to know the total average horsepower developed for the day's run?"



THE NEW MANAGER WANTED TO KNOW HOW MUCH COAL I BURNED PER HORSEPOWER DEVELOPED

I had to admit that he had the situation sized up to perfection and, seeing I was caught "with the goods," admitted I did not know how much coal was required per horsepower-hour in our plant. I even went a little further and admitted I did not know how much coal was being burned each day.

Wood looked surprised, and I felt ashamed.

The manager recovered first and said, "Surely you are aware of the importance of knowing how much coal is being consumed. Coal is the most expensive item in power-plant operation, and the man who can save coal is the man who is appreciated."

I could not very well say anything, seeing I was "in up to my eyes," and so wisely remained silent.

"Let us take this plant just as it is," he continued. "Here is an engine having a cylinder 48x72 and the engine runs at what speed?"

"Fifty-five," I almost yelled, glad that I could answer at least one question intelligently.

"Let's see, that would be about 2100 horsepower at one-quarter cutoff with your steam pressure of 110 pounds per square inch."

I nodded my head, although I did not know what the horsepower of the engine would figure at one-quarter cutoff. I had been told that the engine had a capacity of 2400, but had never taken the trouble to find out what horsepower was being developed at various ranges of cutoff.

"Suppose that the engine required four pounds of coal for each horsepower developed, that would mean a coal consumption of 8400 pounds per hour, or 84,000 pounds each working day, not figuring in the coal necessary for starting and banking fires."

I admitted that he was right, but said no more, not knowing at what he was driving.

"Now, as coal costs $4.10 a ton, the coal bill would amount to $172.20 a day that the company must pay for that one item alone."

I nodded my head for I had figured the thing out, being mentally quick at figures when I wanted to be.

"I suppose you have heard the saying that a good fireman can more than save his wages. Well, that applies to an engineer as well. If an engineer can save a pound of coal, or even a fraction of a pound for each horsepower developed by the engine, it won't require much of a saving to more than equal his wages, in, say, a plant like this."

I nodded my head again, and as I began to see the drift of his talk, cold chills chased each other up my spine.

"Now," went on Wood, "if this plant requires four pounds of coal for each horsepower developed, which I very much doubt, and you reduce the coal consumption one pound for each horsepower per hour, there would be a saving of 2100 pounds of coal each day, over a ton, a saving of $4.30 a day. That's about as much as you get, isn't it?"

"More," I answered, "I get $3.50 a day," and to tell the truth I felt ashamed to think that I might possibly be wasting more coal value than the amount of my pay.

"Of course," Wood continued, "if the saving from four to three pounds amounts to so much, reducing the coal consumption from three to two pounds, which should be obtained from the equipment we have here, would save an equal amount. Of course, it is my business to run this company as economically as it is possible, and at the same time keep it in a strong operating condition. The steam end is one of the great expenses that shows no return for the money invested.

"Now, in order to get down to business, I want a report each week showing the indicated horsepower at some stated hour, about the middle of the forenoon of each day for one week and the same for the afternoon of each day of the following week. In that way a good average of the horsepower output can be had.

"It would not be a bad idea for you to find out just how much it is costing to develop a horsepower. Then the amount of the coal used each day should go on that report, and any other item that would be of value in checking the average daily performance of the plant."

Wood leisurely arose and, after stretching himself, walked out of the engine room. His manner indicated that he had no further concern about the plant, but I knew better. I had sense enough to know that his friendly talk on coal was a mild warning for me to look after that end of my work.

After Wood had disappeared I sat down and thought things over, and after due meditation began to cuss myself for a blank fool. I had allowed a good chance to make good go by. I knew how to indicate an engine, how to figure a diagram, how to set the valves, but had never had the gumption to take enough cards to get the average horsepower output of the engine for a week, and weigh the coal consumed in order to know how much coal was used in producing a horsepower.

If I had done these things and gone to the new manager before he came to me, and told him how much coal was being used per horsepower developed I

would have had a good start in making good.

If I had looked into this matter—had known that each horsepower demanded a consumption of three pounds of coal—knowing that the engine was in good condition, I would have been in a position to have said, "The coal we have is not as good as it should be. There is altogether too much ash and not enough heat units in it. We can make a saving if we insist on getting a better grade of coal."

But I had not done these things. I was as thousands of other engineers were then and are now, content to start and stop the engine and draw the weekly pay. I knew how to do these things, but did not know enough to realize that the management of any company appreciates the efforts anyone in its employ may make to reduce the running expenses.

Of course, after the talk with Wood I indicated the engine as suggested and weighed my coal, and, although I knew it was the proper thing to do, I felt a resentment against his coming into my engine room and suggesting methods of operation different from what I had been accustomed to follow.

But after a few days this feeling gradually wore away and I consoled myself with the thought that I was not the only engineer who did not know how much coal was being burned per horsepower developed.

The more I thought along that line the more satisfied I felt with myself, and, as a result, instead of profiting by my little experience with Manager Wood, I began to dose again instead of looking around for another opportunity to make good.

A Boiler Efficiency of 82.36 per Cent.

What are probably the largest plants ever built for construction purposes are two installations aggregating 5000 boiler horsepower which are at present in service at the Gatun and Miraflores handling plants at the Panama canal. These plants generate electricity for the operation of the cableways by which the concrete is handled and after the canal is completed it is planned to maintain the plants as an auxiliary to the permanent operation of the locks.

At each plant there are six Keeler water-tube boilers arranged in batteries of two each. They are erected in special steel casings entirely inclosing the front, sides and rear, these casings being lined with 2 inches of magnesia block and 9 inches of firebrick.

The Government specifications for these boilers required an evaporative efficiency of 65 per cent. when evaporating 17,000 pounds of water from a temperature of 210 degrees Fahrenheit at a gage pressure of 205 pounds absolute and 150 degrees of superheat. There was a further provision for a bonus of $1000 for each per cent. efficiency above the contract requirements of 65 per cent. The acceptance test on one of the boilers, the results of which are herein given, was conducted by Government engineers, without the assistance or presence of firemen, engineers or representatives of the manufacturers, and, therefore, may be considered as impartial. Extreme care was taken to have all the instruments standardized and to have every item of the test as nearly correct as possible. Two white firemen were employed throughout the test, relieving each other every thirty minutes. The coal used was weighed on platform scales at the front of the boiler and samples were taken from each wheelbarrow load. The feed water was pumped direct from the heater into weighing tanks, located over the heater, and discharged into two small tanks on the floor of the turbine room, from which it was pumped direct to the boiler under test. Owing to the condition of the Orsat apparatus it was impossible to determine the percentage of carbon monoxide and nitrogen present in the flue gases. The high percentage of CO_2 was obtained by close supervision of the draft, which was regulated according to the changes in load. The results of the test on the basis of contract requirements showed an efficiency of 82.36 per cent. Each plant was accepted and the efficiency bonus paid immediately after the completion of the official tests. The report of the test at the Miraflores plant is as follows:

For construction purposes at the Panama canal there are installed twelve Keeler boilers aggregating a total of 5000 horsepower. During the government acceptance tests, one of these boilers showed an efficiency of 82.36 per cent. and an economic evaporation of 13.10 pounds of water from and at 212 degrees per pound of combustible.

BOILER TEST—MIRAFLORES POWER PLANT.

Kind of fuel	Pocahontas coal.
Method of starting and stopping test	Alternate
Duration of trial	8 hours

DIMENSIONS AND PROPORTIONS OF BOILER.

Grate surface, square feet	64.965
Height of furnace 45.25 front, 39 inches rear.	
Approximate width of air spaces in grate, inches	0.625
Proportion of air space to whole grate surface, per cent	46.87
Water-heating surface, square feet	4079.408
Superheating surface, sqare feet	1084.50
Ratio of water-heating surfacc to grate surface	62.7 to 1

AVERAGE PRESSURES.

Steam pressure at boiler, gage	188.06
Draft below damper, inches of water	0.412
Draft in furnace, inches of water	0.306
Draft in ashpit, inches of water	0.062

AVERAGE TEMPERATURES

Steam from superheater, degrees Fahrenheit	481.79
Degrees of superheat	98.79
Feed water entering boiler, degrees Fahrenheit	183.57
Escaping gases, degrees Fahrenheit	473.62

FUEL.

Size and condition	Run-of-mine.
Dry coal consumed per hour, pounds	1,540.21
Combustible consumed per hour, pounds	1,500.64
Dry coal per square foot of grate surface per hour, pounds	18.966
Combustible per square foot of water-heating surface per hour, pounds	2.942

CALORIFIC VALUE OF FUEL.

Calorific value, by Parr calorimeter, per pound of dry coal, B.t.u.	15,292
Calorific value, by Parr calorimeter, per pound of combustible, B.t.u.	15,683.4

ECONOMIC RESULTS.

Water apparently evaporated under actual conditions, per pound of coal as fired, pounds	10.30
Equivalent evaporation from and at 212 degrees Fahrenheit per pound of coal as fired, pounds	12.53
Equivalent evaporation from and at 212 degrees Fahrenheit per pound of dry coal, pounds	12.76
Equivalent evaporation fromand at 212 degrees Fahrenheit per pound of combustible, pounds	13.10
Equivalent evaporation per hour from and at 212 degrees Fahrenheit per square foot of water-heating surface, pounds	4.82

HORSEPOWER.

Horsepower developed	569.97
Builders' rated horsepower	400
Percentage of builders' rated horsepower developed	142.49

EFFICIENCY.

Efficiency of the boiler, including the grate; heat absorbed by the boiler, per pound of dry coal, divided by the heat value of one pound of dry coal (feed water at 210 degrees Fahrenheit and steam pressure 190 pounds gage) per cent	82.36

COST OF EVAPORATION.

Cost of coal, per ton of 2240 pounds, delivered to boiler room	$4.25
Cost of fuel for evaporating 1000 pounds of water under observed conditions	$0.1841
Cost of fuel for evaporating 1000 pounds of water from and at 212 degrees Fahrenheit	$0.1513

METHOD OF FIRING.

Kind of firing	Spreading
Average thickness of fire, inches	7.666
Average interval between firings during time when fires are in normal condition, minutes	5.23
Average interval between leveling, minutes	26.38

ANALYSES OF DRY GASES.

Carbon dioxide (CO_2), per cent	12.458
Oxygen (O), per cent	2.747

An Engine Built Up of Sections

By A. B. Gould

To meet the demand for a steam engine in a locality which was accessible only by mule pack, a 14x42-inch engine was constructed of sections, none of which weighs over 350 pounds. The engine complete consists of over one thousand pieces.

Occasionally it is necessary to develop power at points remote from any railway, and to which it is not possible to build a wagon road for the purpose of transporting material. In such cases everything must be carried on the backs of mules, over hills and along trails where in some instances it is scarcely possible for even a mule to find a foothold. Machinery to be thus transported must be especially constructed so that when taken apart for shipment, no one piece will be of greater weight than can be easily carried by a mule. This often necessitates a departure from the usual types of construction, as the process of sectionalization sometimes introduces unusual conditions that must be met by unusual construction. Some kinds of machinery are adaptable to the cutting-up process, while others present difficulties that tax the ingenuity of the designer. As an illustration of how a Corliss engine was thus treated, the following description is given.

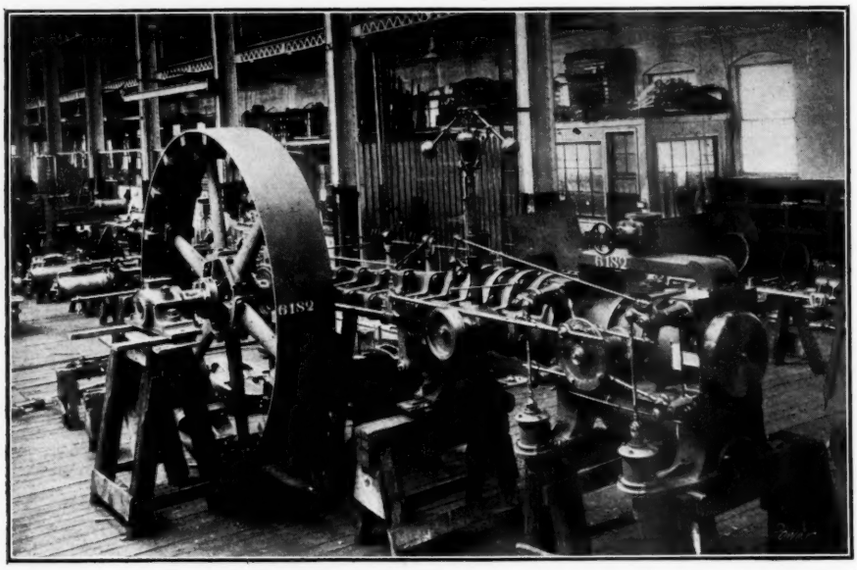


FIG. 1. ENGINE ASSEMBLED IN SHOP

This engine, which is 14x36 inches and runs at 100 revolutions per minute, was built by the Wellman-Seaver-Morgan Company and was recently shipped from its Akron works to the Cia Minera Jesus Maria y Anexas in Mexico.

It may be seen from the illustrations that the engine is built with a girder frame. Owing to the class of work required, it was possible to use this type of frame, which was more easily sectionalized than any other. In this case a pattern for a solid 14x42-inch engine frame was changed by adding bolting flanges at the proper intervals and putting in splitting cores to break up the casting into pieces. These splitting cores were made thick enough to allow for the removal of sufficient metal to shorten the frame to the proper length for a 36-inch stroke. The joints in the frame were faced and the sections held together by bolts to insure the proper alinement of all the pieces. The pillow-block end is separate from the frame, and is divided into four pieces. The bearing is made up in the usual quarter-box manner, providing for all necessary adjustments, and the center of the frame rests on a leg which forms a support for the rocker arm.

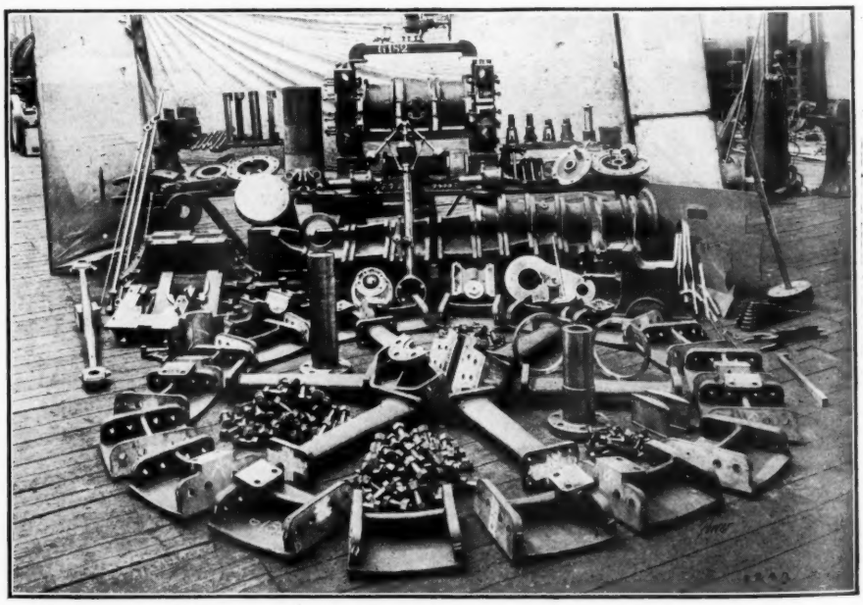


FIG. 2. ENGINE TAKEN APART FOR SHIPMENT

The valve gear is of the standard releasing type, operated by a single wrist-plate attached to the side of the cylinder, and the steam valves are closed by noiselessly operating dashpots which contribute much to quiet running.

The flywheel is 10 feet in diameter and has a 20-inch crowned face. It is made in twenty-six pieces, securely bolted together with fitted bolts where necessary to preserve the proper relation of the various parts. The arms and rim are of cast iron, and the hub of cast steel, the latter being in two pieces bolted together, and still further secured by two steel rings shrunk on the outside of the flanges where bolted to the shaft. The shaft is in two pieces, one bolted to either side of the flywheel hub, making the hub part of the shaft. These two sections of the shaft are annealed-steel castings with a hole cored through each. This construction allows a considerable saving in weight, with but a small loss in strength.

The most interesting part of the entire engine is the cylinder. The barrel is in four pieces with the valve chests separate. The three outside pieces of the body are doweled and bolted together and contain the cast-iron bushing, forming the cylinder proper through which the piston travels; this construction eliminates any joints over which the piston must

travel. To each end of the barrel are bolted the chests containing the steam and exhaust valves. These chests are divided horizontally at the center line of the cylinder. The steam chest is a separate casting and is bolted to the valve chest directly over the steam valve, while the exhaust from each end of the cylinder is carried down through the cylinder leg into the exhaust chest and from there to the exhaust pipe. The whole cylinder is lagged with magnesia and is covered by a steel jacket.

The other parts of the engine are of standard construction, and did not require to be made in sections. No part as shipped exceeded 350 pounds in weight, this being the limit imposed.

Fig. 1 shows the general appearance of the engine, minus the cylinder jacket, as it stood on the erecting floor, and Fig. 2 is a view taken after it had been dismantled and was ready for shipment. It is interesting to note that there are over one thousand pieces in this engine, all of which were successfully transported and erected with the loss of only a piston ring.

Charging a Refrigerating System

By F. E. Matthews

The author gives explicit directions for charging and a number of tables devoted to cubical contents of pipes and weights of gas at different pressures, are presented to assist in estimating the amount of charge required.

After it has been ascertained that the system is perfectly tight, the air and the ammonia should be allowed to escape, and the whole system should be pumped down to a vacuum. The expansion valve is first closed, then the valve on the drum of ammonia connected with the charging valve is opened. The liquid is allowed to flow through the slightly opened expansion valve into the system under the pressure of the gas above the liquid in the ammonia drum, just as water is forced out of a boiler by the pressure of the steam. It was suggested that the system be pumped out to a vacuum, not because a vacuum is necessary to draw the liquid around into the system, but rather to get rid of as much of the air as possible. While the production of a lower pressure within the refrigerating system than that of the atmosphere without undoubtedly hastens the operation of charging, there will be a tendency to draw air or water into the system, so that a vacuum should never be pumped until it has been demonstrated beyond doubt that the system contains no leaks. Small leaks into the system are not readily detected, and much more trouble can be made by what water a small leak will let into a system than by the amount of ammonia the same leak will let out.

Even pumping a vacuum does not insure the expulsion of all the air. The remainder may be allowed to stay in the system until displaced by purging at the condensers, or a large percentage of it can be driven out of the system during the process of purging by the judicious manipulation of the ammonia. If the ammonia be admitted very slowly at one end of a long run of pipe, it will drive the air before it without mixing with it to any great extent, and, if at the other end of the pipe line a valve be opened or a flange union be cracked after sufficient ammonia has been admitted to produce a pressure above that of the atmosphere, the air can be allowed to escape until it contains too large a percentage of ammonia, when the opening is closed. That the liquid ammonia will pass from the shipping drum into the system without the necessity of pumping is evident.

If the engine-room temperature be 80 degrees Fahrenheit, for example, the pressure in the drums will be 140 pounds. If the drums are exposed to the sun, the temperature may rise much higher than that of the surrounding air and dangerous pressures may result. It is accordingly advisable to store ammonia drums in a cool place. The shipping drums are designed to carry any reasonable pressures but there is a remote possibility that the drum may be filled too full. In this case, since there is not sufficient vapor space to take care of the expansion of the liquid as the temperature increases, and since liquids are practically incompressible, there is no limit to the amount of pressure that may be produced except that of the ultimate strength of the drum. There is the same danger in tightly closing the valves on all the outlets to the liquid receivers when it is not definitely known that they are not completely filled with liquid. Explosions due to such causes are second only to boiler explosions in their disastrous results.

It is obvious that the vapor generated in the drum will drive the liquid out into the system so long as the temperature of the liquid is such as to produce gas to keep the pressure higher than that in the system. If, for example, the system is operating under 15 pounds gage back pressure, 16 pounds vapor pressure in the drum would suffice to expel the liquid. The temperature of the liquid corresponding to a pressure of 16 pounds is about zero degree Fahrenheit. From this it will be seen that the only disadvantage of charging against back pressure is that the liquid will not flow so rapidly into the system because of the decreased difference in pressure. The slightest reduction in pressure within the drum, due to a removal of part of the liquid, causes the ammonia to boil more vigorously, generating more vapor to fill the increasing space above the liquid. The temperature at which the liquid boils gradually drops, however, until at a pressure of about 47 pounds, which corresponds to a temperature of a little less than 32 degrees Fahrenheit, the pipe leading from the drums will become sufficiently cold to precipitate and congeal atmospheric moisture, and is said to "frost." The melting of this frost indicates that there is no more ammonia passing through the pipe, and the valves can accordingly be closed, the empty drum removed and a full drum connected.

So far as the production of cold is concerned, there need be only sufficient liquid refrigerant to insure a solid stream at the expansion valves, so that no gas may enter the expansion coils at any time. This condition can be readily recognized by the intermittent hissing sound produced by the passage of quantities of liquid and gas. The condition in which there is just enough liquid to give a solid flow at the expansion valve is the minimum charge that can be economically employed without loss of both capacity and efficiency.

To provide for unforeseen contingencies, such as losses of ammonia through leaks, temporary trapping of liquid in low parts of the system, etc., it is always expedient to have the liquid charge somewhat in excess of this amount, a kind of credit balance in the bank to guard against the embarrassment of overdrawing one's account if collections do not come in from the freezer, expansion coils, condensers, etc., as expected.

Increasing the charge of anhydrous ammonia above that actually required to insure an uninterrupted flow at the expansion valve, will work no harm to the system up to the point where the liquid fills the receiver and begins to encroach upon the condensing surface. To be sure, the more liquid lying in the compression side of the system under the usual conditions of operation, the less space there will be for the storing of additional anhydrous ammonia should it become necessary to "pump out" any or all of the coils of pipe in the low-pressure side. The additional ammonia occasions an additional investment, but in the majority of plants it is expedient to carry a small

stock of liquid to provide for contingencies; and aside from the rather remote possibility of an accident that would result in the loss of the entire charge, it is better to have the ammonia in use in the system than lying idle in shipping cylinders.

An overcharge of ammonia in a system can usually be detected in two different ways. Since the condensed liquid is usually several degrees colder than the uncondensed gas, the parts of the compression side that contain liquid ammonia, whether the liquid receiver connecting piping or pipes of the condenser, can usually be determined by their temperatures. Assuming that the plant is overcharged to such an extent that the liquid in the compression side begins to fill up the condenser, thus encroaching on the available condenser surface, a material rise in head pressure over that usually observed when running under similar conditions of speed, back pressure and water supply with only sufficient liquid in the system to insure a solid flow through the expansion valve would be expected, although a slight increase in head pressure observed on increasing the charge should not be accepted as positive proof of overcharging.

TABLE 1. RELATION OF CUBICAL CONTENTS TO RUNNING FEET IN PIPES OF VARIOUS SIZES.

Size of Pipe, Inches.	Running Foot per Cubic Foot of Contents.	Contents in Cubic Feet per 100 Running Feet.
¾	270.00	0.370
1	166.90	0.599
1¼	96.25	1.038
1½	70.65	1.415
2	42.36	2.360

TABLE 2. WEIGHTS OF AMMONIA VAPORS AT DIFFERENT GAGE PRESSURES.

Ammonia Gage Pressure.	Weight of 1 Cubic Foot of Vapor, Lb.	Ammonia Gage Pressure.	Weight of 1 Cubic Foot of Vapor, Lb.
0	0.0566	80	0.3304
10	0.0941	90	0.3617
20	0.1269	100	0.3939
30	0.1611	125	0.4766
40	0.1955	150	0.5566
50	0.2292	175	0.6340
60	0.2641	200	0.7188
70	0.2965	...	.....

There can be no fixed rule by which to determine the amount of ammonia required for a direct-expansion refrigerating system. For systems not including sharp freezers, the only accurate means is to calculate the amount of the charge, taking as a starting point the amount of pipe to be filled with the refrigerant in both the high- and the low-pressure sides of the system. The accompanying tables, showing cubical contents of pipes and weights of gas at different pressures, will be found convenient when calculating the amount of ammonia required to charge the system.

The number of hundreds of running feet of pipe in the system having been determined, the cubic feet contained in it may be found from Table 1. The amount of ammonia necessary may be ascertained by multiplying the cubical contents by

TABLE 3. ANHYDROUS AMMONIA REQUIRED FOR THE COMPRESSION SIDE OF REFRIGERATING PLANTS.

Tons of Refrigeration.	Pounds of Ammonia.	Tons of Refrigeration.	Pounds of Ammonia.
5	110	75	375
10	150	100	440
15	185	150	510
20	230	175	570
25	245	200	620
30	270	225	675
35	290	250	725
40	300	300	840
45	325	400	1040
50	350	500	1215

the weight of gas per cubic foot corresponding to the pressure to be carried on the pipes when the system is in operation. The weight of ammonia vapor required to fill both high- and low-pressure sides of the system may be determined in this way, in addition to which a liberal margin should be allowed for reserve liquid in the receiver, evaporating liquid in the expansion coils and condensing liquid in the condenser.

Where sharp freezers are in service a much larger amount of liquid will be required to charge the low-pressure side, the extra charge increasing very rapidly with decreasing pressures.

When the refrigerating machinery is to be operated under average conditions an ammonia charge figured according to the following table will be in line with commercial practice.

TABLE 4. ANHYDROUS AMMONIA REQUIRED PER 100 RUNNING FEET OF PIPE—EXPANSION SIDE.

Refrigerating Plants. Direct Expansion and Brine Cooling Coils.	Size of Pipe.	Ice Plants. Expansion Coils for Can and Plate Use.
14 pounds.	1 inch.	8 pounds.
18 pounds.	1¼ inches.	11 pounds.
20 pounds.	1½ inches.	12 pounds.
25 pounds.	2 inches.	15 pounds.

In ice plants the amount of expansion surface per ton is more nearly a constant than in direct-expansion refrigerating plants. Since different sizes of expansion piping are used by different builders, the expansion surface does not always bear a fixed relation to the space to be filled with ammonia vapor. The following ammonia charges for ice-making plants may be considered in line with average practice.

TABLE 5. AMMONIA REQUIRED FOR ICE MAKING PLANTS.

Tons of ice per 24 hours.....	5	10	15	25	50	100
Pounds ammonia.	100	250	500	1000	2000	4000

The amounts given in Table 5 are for the total number of pounds required to charge both high- and low-pressure sides of the ice-making systems in question, while those in Table 3, are those required for the compression side only.

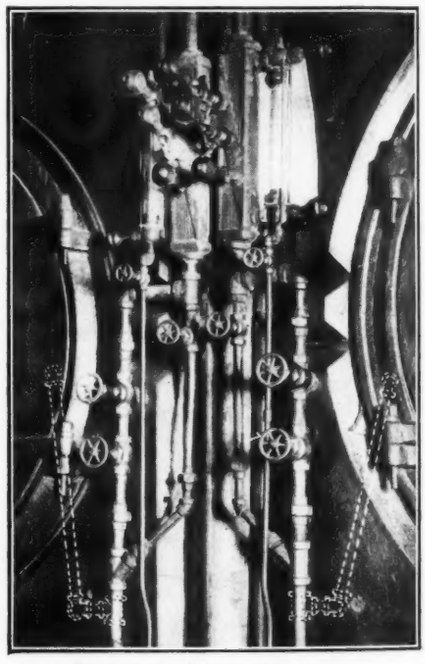
The amount of sodium chloride (NaCl) required to make brine for an ice tank of given capacity is also subject to wide variations on account of the diversity of designs used by the different builders.

A good general rule is to allow 15 pounds of salt per cubic foot of brine actually required to fill the tank when the cans are in place. Another rough rule is to allow two-thirds ton of salt per ton of ice-making capacity of tank per 24 hours. When calcium chloride (CaCl) is employed, some authorities estimate the amount required at one ton, per ton of ice-making capacity.

Trouble with Water Column

A trouble that many engineers have encountered in boiler-room practice is the failure of water to return to the water glass and column after they have been blown out.

In one instance the water columns on two boilers were originally piped as shown in the accompanying illustration



PIPING OF WATER COLUMNS

by the dotted sections of piping. When the water column was blown down, the water failed to show at normal water level for about fifteen minutes.

When the top try cock on the water column was opened, the water would shoot to the top of the glass and stay there. When the bottom try cock was opened, the water would disappear for a period of from ten to fifteen minutes.

A boilermaker, several experts and some good engineers could give no explanation as to why the column should act in this way, and failed to suggest a remedy. The engineer finally made the connections between the boilers, as shown in the original lines of the photograph and the trouble disappeared.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Care of Rotary Converter Commutators

By W. C. Motley, Jr.

I have recently read several articles in Power dealing with rotary converters, but a few more general remarks on their care may be of interest to your readers. Many operators have the idea that the commutators in these machines cannot be kept in good order but, while I am aware that they require perhaps more attention than commutators on ordinary generators, the desired polish can be attained by close attention to details.

One very often finds commutators having very hard mica in them; the result is that when the copper wears away a little the brushes chatter and spark due to the poor contact caused by the high mica strips. The ordinary sandpapering is not effective in curing the trouble, and finally the superintendent orders the commutator turned down. This is wasteful and usually unnecessary, as the operator can keep the machine in order if he will follow the plan suggested.

Coarse and fine sandpaper are necessary and it is a good plan to rub the commutator with an oily rag, so that the dust will collect on the paper instead of in the machine. The commutator should be thoroughly rubbed with the coarse sandpaper until it is free of burnt spots. Then the fine paper should be used until the surface is bright and clean. In both of these scourings, the sandpaper block should be applied with considerable pressure. After the surface is well polished, apply the fine sandpaper again, *but do not press too hard this time*. This will just touch the high spots and will have the same effect as holding a lathe tool close enough to cut the mica without losing any copper.

Many operators advise cutting the mica away between bars, leaving a shallow groove between the faces of adjacent bars. No doubt this gives some good results, but there is a chance of dirt lodging in these spaces and making it possible for the machine to "flash over." It also puts the machine out of commission while the job is being done, which is objectionable in cases where the machines carry a heavy load.

For cleaning a commutator an oily rag is preferable to gasolene; gasolene sometimes leaves a sticky surface which causes the brushes to glaze and chatter. Be sure, however, to get all the oil off, as too much oil will cause trouble when the machine is running.

The direct-current brushes also require considerable attention and it is important that they be kept clean at all times; it is unwise to let them go until they become gummy. They should make perfect contact with the commutator and no cracked or "honeycombed" brushes should be tolerated. In sandpapering sets having a dozen or so brushes, it will be found much easier if the paper is moved *parallel* to the commutator bars and not in the usual direction, provided the design of the machine will allow it, as there is then no danger of rounding the toes and heels of the brushes. Of course, sandpapering each brush individually is much better than doing the whole set at once, but time is often too limited for that.

I have heard it said that commutators should be lubricated at stated times, but I do not see why this is so, as we all know that grease is a trouble maker. It is much better to let a machine alone as long as it runs quietly, greasing it only when absolutely necessary and then using as little as possible. The selection of a lubricant for commutators is also important. Some people use compounds and some use wax; others use dynamo oil or vaseline. The objection to most compounds and waxes is that if the machine runs cool they gum the brushes. Oil or vaseline will seldom do this and though they must be applied oftener, they usually give better results.

The alternating-current side of a rotary converter does not usually give much trouble. The only one worth mentioning is that the brushes sometimes cut due to stiff brushes, too much pressure or lack of lubricant. The cures for these troubles are obvious. I once operated some 400-kilowatt Stanley alternators the rings and brushes of which had regular saw teeth in them due to neglect. Rather than have them turned down in a lathe, I filed the teeth off the brushes, set them tight on the rings and did not use any lubricant. Naturally, the teeth in the rings began to wear away and as I kept the brushes filed smooth and square the rings soon got smooth also, and took on a fine polish when lubricated. The rings in this case were made of copper and lent themselves well to the treatment. Had they been of other material, I might not have been so successful.

Some superintendents are beginning to realize that better results can be obtained if all commutators are placed under the care of one man. In stations where shifts are changed from time to time and different men work on the machines, there is not the same interest taken as there would be by a man who thoroughly understands the work and is responsible for it. The old saying, "Too many cooks spoil the soup" is not out of date yet.

Direct Current Motors and Their Connections

By J. M. Row

A great many times when connecting up or making repairs on a motor, a little familiarity with the elementary principles and operating characteristics of the machines would save engineers lots of time and worry.

In dealing with motors there are three facts which should not be lost sight of; first, if an electric current is passed through a conductor which lies across a magnetic field, the conductor will try to move. Second, if a conductor moves across a magnetic field, "cutting" the lines of force, an electromotive force will be generated in the wire and this will tend to produce a current in such a direction as to oppose the motion of the conductor. Third, if an electric current be passed around a coil of wire having a soft iron core, the core becomes a magnet. Later on the significance of these three principles will appear.

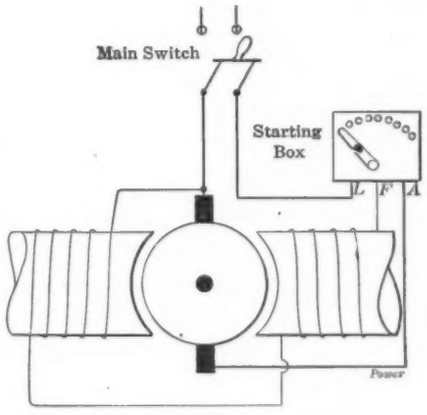


Fig. 1. Shunt-wound Motor

In referring to motors in this article only those which operate on direct-current circuits are meant. They are divided

into four general classes: shunt-wound, series-wound, cumulative-compound and differential-compound. The shunt-wound machine is the one most commonly used where practically constant speed is desired. Fig. 1 shows how it is connected up. To put it in service, the main switch is first closed, then the resistance in the starting box is slowly cut out by turning the handle to the right, allowing the motor to come up to speed slowly. It will

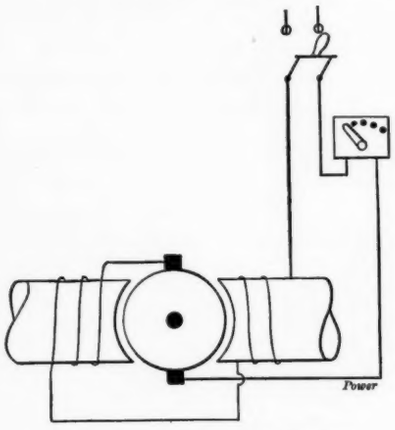


FIG. 2. SERIES-WOUND MOTOR

be noticed that the current divides at the positive brush, part flowing through the field winding and the other part, which is much the greater, passing through the armature. The armature conductors, being located in a magnetic field and carrying an electric current, will

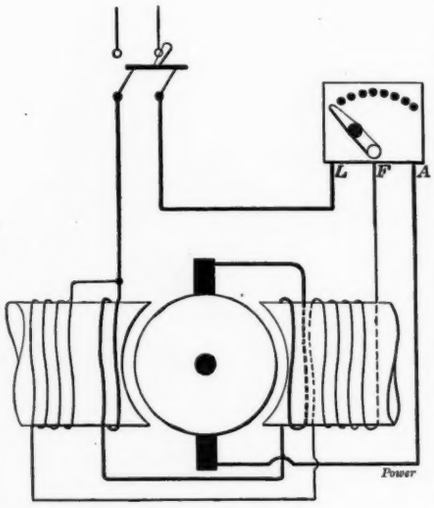


FIG. 3. COMPOUND MOTOR

move, causing the armature to revolve, in accordance with the first of the principles previously stated. The rotation of the armature, moving the conductors across the field, will generate an electromotive force in them in such a direction as to oppose the flow of current. This is called the counter electromotive force of the machine, and it varies with the speed; that is to say, the greater the speed the greater the counter e.m.f. Also, since the counter e.m.f. opposes the flow of current, the greater it becomes the less will be the current. This makes the motor automatic in its operation, for if the load be increased the armature begins to slow down, thus decreasing the counter e.m.f. and allowing the current to increase sufficiently to pull the increased load. The speed of this motor may be varied between certain limits by inserting "dead" resistance in either the field or armature circuit, or both. The speed will be increased by increasing the resistance of the field circuit, and decreased by increasing the resistance of the armature circuit.*

The series-wound motor, represented diagrammatically in Fig. 2, is used where the motor is to be repeatedly started and stopped and starts with a heavy load, such as driving freight elevators, street cars, etc. It is well adapted for this kind of work because of its large starting torque. But where constant or nearly constant speed is desired this type of motor is practically useless, for every slight change of load will make a very great change in the speed. Moreover, should the load be thrown off, the motor would immediately race to such a high speed as to completely destroy it. In this respect it acts just like an engine when the governor fails to operate.

It is often necessary to have a motor start with a large torque and at the same time retain the constant speed characteristic of the shunt-wound machine. To meet this condition the ordinary compound-wound, or cumulative-compound motor has been designed. As indicated in Fig. 3, it has two field windings, one in series and one in shunt to the armature; these give it a characteristic combining those of both the series and shunt-wound motors. A great deal more care must be exercised in connecting up this machine. First make sure that the two windings are not interchanged. They can easily be identified by examining the wires leading into the coils; the one composed of the larger wire is the series winding. In connecting up it is best to connect the series field winding first, and test the machine to see if the armature runs in the right direction; if it does not, the connections of the field winding must be reversed. Then connect up the shunt field winding and, with a temporary "jumper" across the terminals of the series windings, test again for direction of rotation, reversing the terminals of the shunt winding this time if the armature runs the wrong way. Remove the jumper and the motor is ready for regular work.

It often happens, usually by carelessness, that in connecting up a compound-wound motor the series winding is connected in opposition to the shunt winding. This makes the machine a differential compound motor. This fault, when present, will make itself known by the motor stopping and reversing its direction of rotation (providing the circuit-breaker does not open) every time the load is increased to a certain amount; or even sometimes while the starting resistance is being cut out.

*See article on "Varying the Speed of an Electric Motor," by Cecil P. Poole, page 735, November 2, 1909, for fuller information on this point.

Central Station Growth

That the central-station industry is growing rapidly is made evident by a recent special report of the Census Bureau at Washington. The report covers the period between 1902 and 1907, during which the central-station capacity in this country was more than doubled.

In 1902 there were 3620 plants in operation, representing the investment of $504,740,000 and having a total capacity of nearly 2,000,000 horsepower. In 1907 there were 4714 plants, representing the investment of about $1,100,000,000 and having a capacity of slightly more than 4,000,000 horsepower. During the five years, therefore, the capacity increased about 120 per cent. and the total cost about 117 per cent., while the increase in the number of plants was only 30 per cent. The total output in 1907 was approximately 5,862,000,000 kilowatt-hours, and in 1902 it was slightly over 2,507,000,000; this is a gain of about 134 per cent.

These figures serve to illustrate the influence of the continual increase in the size of the units and the accompanying decrease in the space required per unit of capacity. Had the increase in the number of the stations or even the increase in the sizes of the stations kept pace with that of capacity, the increase in cost would in all probability have far exceeded the gain in capacity.

The report divides the central-station employees into two classes, the one composed of salaried officials, clerks, etc., and the other of wage earners. The number of those in the first class increased by 86 per cent., while that of the second group gained only 49 per cent. Here, too, the increase in the size of the unit has had an influence. The average salary rose about 11 per cent. while the average wage rate rose about 4½ per cent.

The number of reciprocating steam engines and steam turbines increased from about 6000 to approximately 7200, or 20 per cent., while their total capacity rose from 1,379,941 to 2,627,450 horsepower, an increase of 90 per cent.

The number of gas-engine units increased from 165 to 463, or 181 per cent., while their total capacity rose from about 12,000 to about 56,000 horsepower, or 343 per cent.

In 1902 there were 1390 water wheels in service, while in 1907 there were 2481. Their capacity increased from about 440,000 to nearly 1,350,000 horsepower, which is a gain of about 208 per cent.

On account of the greater adaptability

of the alternating-current generator to central-station requirements, the number of direct-current generators in use diminished from about 7400 to about 5400, or about 27 per cent., while that of the alternating-current machines rose from 5122 to 6808, or 33 per cent.

The foregoing figures do not represent the entire production of electrical energy. To arrive at the aggregate it would be necessary to consider also the electric railways and the isolated electric-light and power plants, not to mention the relatively small-power telephone and telegraph lines, electric police-patrol and fire-alarm systems.

The report states that in 1907, exclusive of the isolated plants, there were upward of thirty thousand individuals, companies, corporations and municipalities which reported the generation or utilization of electric current in what may be termed "commercial enterprise." These industries represent an outstanding capitalization of over $6,000,000,000, of which about $1,333,000,000 is credited to central stations, $3,667,000,000 to electric railways, $800,000,000 to commercial or mutual telephone companies and about $250,000,000 to telegraph companies, the latter item including nearly $33,000,000 of capital stock of wireless companies.

Although central-station statistics of the comparatively few street railways that sold current and that were able to prepare complete separate reports have been included with those for central electric stations, in order that that branch of the electrical industry might be reported as distinctly as possible, the full measure of the growth of the central station is not made evident by the figures given because of the fact that the sale of current for light and power is also largely carried on by numerous street-railway companies which combine central-station business so completely with other activities as to make complete separate reports impossible.

As would naturally be supposed, central stations are concentrated in the most populous States and at points within these States from which the largest portion of the population can be served economically. New York, Pennsylvania, Illinois and Ohio, the four States having the largest populations, contain nearly 30 per cent. of the total population of the United States. In 1907 these four States reported 1296 electric stations, or 27.5 per cent. of the total number in operation during that year. The annual output of these stations amounted, roughly, to 2,500,000,000 kilowatt-hours during 1907, which is 43.6 per cent. of the total. While the proportionate number of central stations and the proportionate population of this group of States were very nearly the same, and less than one-third of the respective totals, their output was nearly one-half of the total.

The Burke Induction Motor

The Burke induction type of alternating-current motor is a very rugged, substantial machine, as may be inferred from an inspection of the accompanying illustrations. The stator or primary member, shown in Fig. 1, comprises the usual slotted core of laminated steel mounted in a massive cast-iron housing and equipped with form-wound and individually insulated coils. The core laminations are keyed to the housing, in order to avoid the use of bolts through the core, which entail eddy-current losses. The slots in the core are partly closed, unless otherwise ordered, because this construction increases the effective airgap area, thereby decreasing the gap reluctance and therefore the magnetizing current; this improves the efficiency slightly and the power factor appreciably. The construction of the stator housing is obvious from the picture.

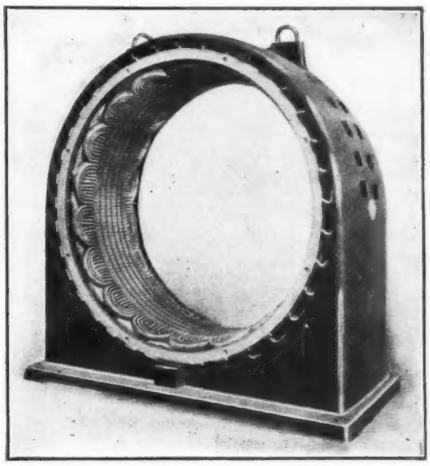


FIG. 1. STATOR OF BURKE MOTOR

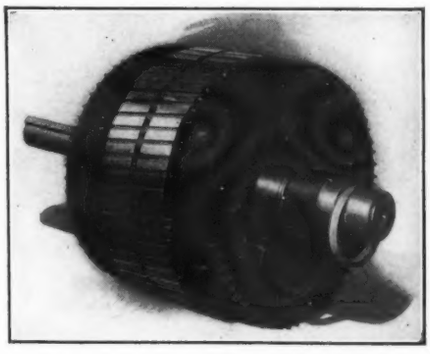


FIG. 2. ROTOR OF BURKE MOTOR

The rotor core disks are assembled with the conductor slots progressively twisted out of line just enough to make the completed slot diagonal, as illustrated in Fig. 2. This reduces the flux pulsations in the airgap, which tends to reduce iron losses and magnetic humming. The rotor conductors of the squirrel-cage type consist of one-piece closed loops punched out of copper sheet in the form shown at the top of Fig. 3 and afterward pulled out into the form shown at the bottom of the same engraving. The two straight "sides" or parallel parts of the loop are separated to a distance approximately equal to the polepitch of the stator, and it is therefore unnecessary to connect the ends of the various coils together; each one forms its own closed circuit. The slip-ring type of rotor is, of course, equipped with a "polar" winding, the terminals of which are connected to collector rings. This form is used for variable-speed service and for conditions requiring high starting torque with moderate starting current.

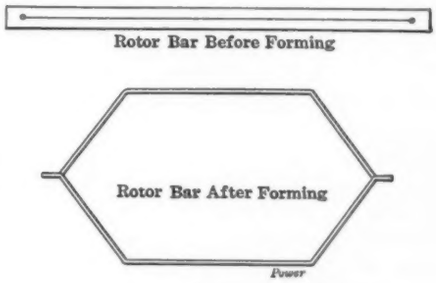


FIG. 3. A ROTOR BAR

The bearings are mounted in the usual circular bonnets or brackets, fitted to the face and bore of the stator housing. The journal sleeves are cast-iron shells with babbitt linings, and the oil rings consist of a number of thin metal disks, as indicated in Fig. 4. This arrangement is very unusual, but it is said to give excellent results. In the bearing from which this picture was taken there were ten of these disks.

Burke motors of 5 horsepower and under are not supplied with starters; those of larger output are regularly supplied with either resistor-type or auto-transformer starters. With the latter type a separate oil switch is supplied.

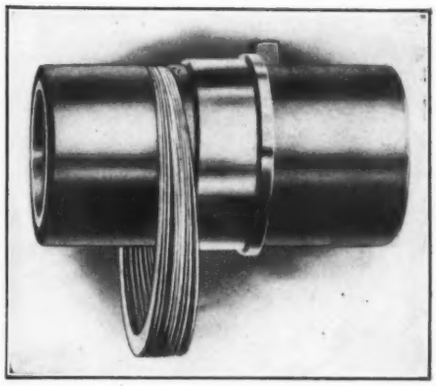


FIG. 4. LAMINATED OIL RING

These machines are built by the Burke Electric Company, Erie, Penn.

Sid Diggles ast Hal ef he knowed why his bilers wuz like a herd uv melk cows. Hal sed he didn't, an' didn't keer a cuss nohow. Sid told 'im 'twas cause they hed so dumd many bags on 'em.

Gas Power Department

Flying Machine Motors

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

The recent aviation "meet" at Belmont park, near New York City, afforded an excellent opportunity for close inspection of the various types of aëroplanes and their power plants. As our readers doubtless know, an aëroplane consists of huge planes or wings of light material mounted on a framework which also carries a two-blade propeller and a gasolene engine to drive it. While flying machines do not come within the regular field of POWER, the engines that propel them are of interest to all of us by reason of their kinship to stationary and marine engines driven by gasolene.

All of the engines used in the Belmont Park contests operated on the four-stroke cycle, and all were of the multiple-cylinder type with single-acting pistons, following rather closely the general lines of automobile practice. The machine used by the Wright brothers departs from this practice less than any

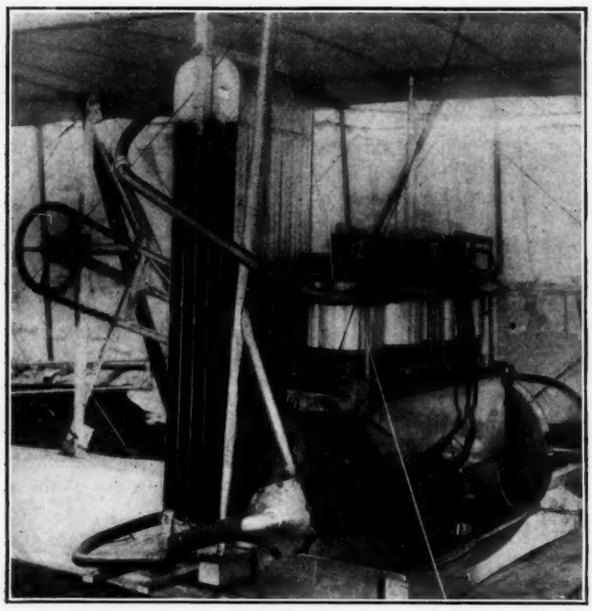


FIG. 1. THE WRIGHT MOTOR

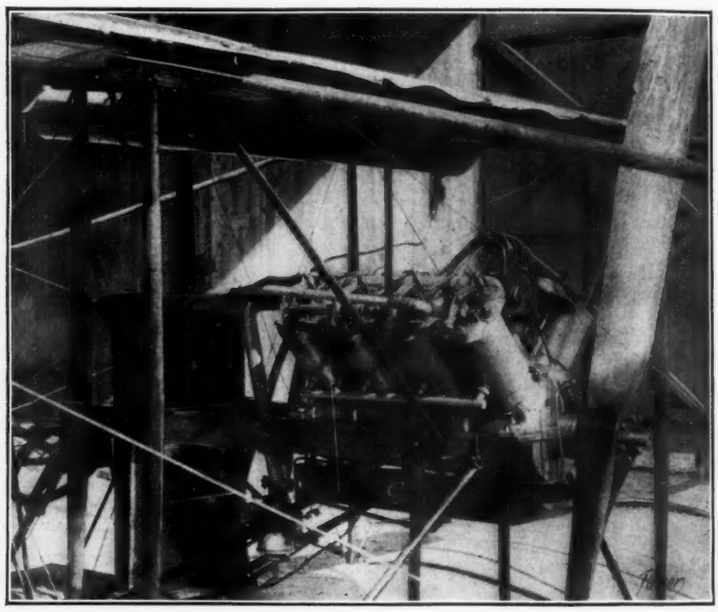


FIG. 2. CURTISS' REGULAR MOTOR

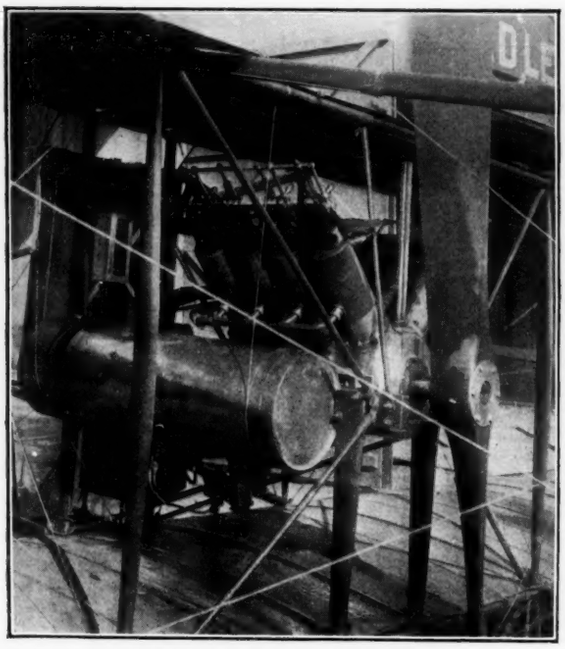


FIG. 3. HAMILTON'S MOTOR

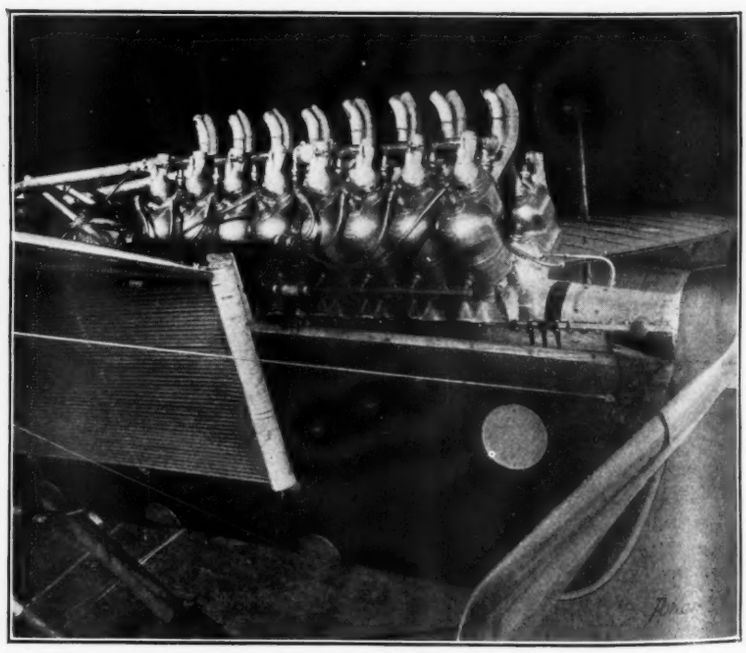


FIG. 4. THE 100-HORSEPOWER ANTOINETTE MOTOR

of the others. This is a four-cylinder engine of the appearance shown in Fig. 1. It embodies no features radically different from the ordinary motor-car engine; its weight has been reduced to the lowest practical point and there are some special details in the arrangement of the ignition magneto; that's about all.

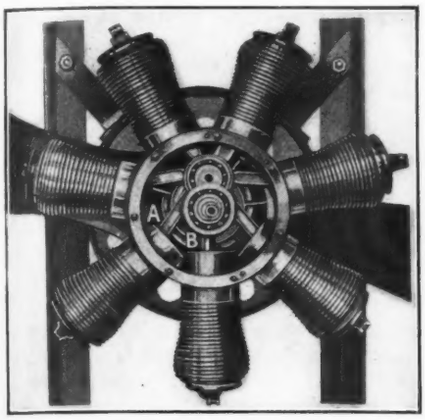


FIG. 5. GNOME ROTATING CYLINDER MOTOR

All of the other engines, excepting the Gnome, were of the V type; that is, the cylinders are divided into two rows set at an acute angle with each other, as shown in Fig. 2, which illustrates the Curtiss motor. The valve gear of the Curtiss engine is unusual in that both the inlet and exhaust valves of each cylinder are opened by one rocker arm. The chief object of this arrangement is to reduce the weight. Only one rocker arm and one push rod are needed for each cylinder. The openings in the cylinders near the heads are exhaust outlets; no exhaust pipe is used—another weight-reducing expedient. The long, slender cylinder with pointed ends, suspended above the engine, is the gasolene tank, from which the fuel flows to the carbureter, located below the rear end of the engine. The ignition magneto is located between the two rows of cylinders at the front end of the engine.

The construction of the engine used by Hamilton (Fig. 3) is similar to that used on the regular Curtiss machine. The differences between them are not of fundamental importance.

The Antoinette engine, illustrated by Fig. 4, is also of the V type, but it embodies the radical feature of injecting the gasolene directly into the intake pipe instead of atomizing it and mixing it with air in a carbureter. One result of this arrangement is that it is extremely difficult to start the motor; another is that the speed cannot be regulated by varying the admission of fuel, except to a very limited extent, because decreasing the quantity of gasolene injected makes the mixture in the cylinder too lean for prompt ignition and increasing it makes the mixture too rich. Most of the speed control is effected by changing the timing of the ignition, which overheats the engine at the lower speeds. The short pipe extending upward from each valve chamber is the intake pipe; the fuel nozzle enters the pipe on the side in a direction almost parallel to the axis of the pipe. The longer pipes projecting upward in pairs and curving backward are the exhaust pipes. The spark plugs are located in the centers of the cylinder heads. The manifold extending alongside the body of the craft, from the sixth cylinder rearward, is one of the radiators for cooling the jacket water; the other one is in the corresponding position on the other side of the machine. The tubes and headers are made of aluminum, to reduce weight.

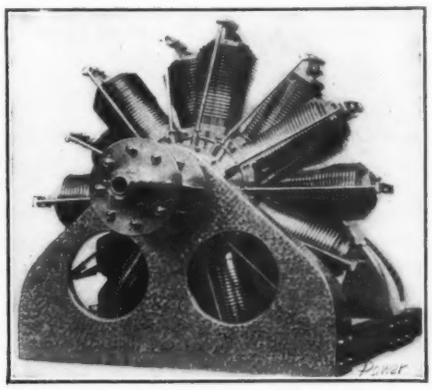


FIG. 6. DOUBLE ROW GNOME MOTOR

The Gnome engine differs from all of the others that were at this meeting in two important particulars: the cylinders revolve about a stationary crank shaft and they are air cooled, instead of being water cooled. Fig. 5 is a view of a seven-cylinder Gnome engine with the crank-case cover removed to show the arrangement of the connecting rods and crank. There is only one crank and that is located above the shaft; consequently, the lowermost piston always projects beyond the end of its cylinder, as shown at *B*. The engine is built with single and double ranks of cylinders; the double-rank engine has fourteen cylinders, seven in each rank, as shown in Fig. 6, and two cranks located 180 degrees apart. This type was used by Grahame-White at Belmont park. Fig. 7 is a view of the motor in the Blériot machine, and Fig. 8 is a side view, showing that the cylinders in the two ranks are staggered with respect to each other. This makes the engine shorter, and therefore the weight less, than it would be with the cylinders in line fore-and-aft, and it also allows the air to get to the rear cylinders more effectively.

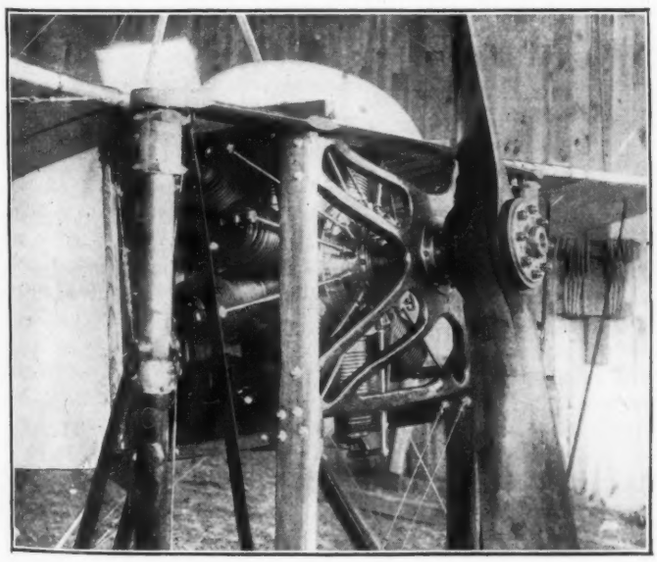


FIG. 7. GRAHAME-WHITE'S 100-HORSEPOWER GNOME

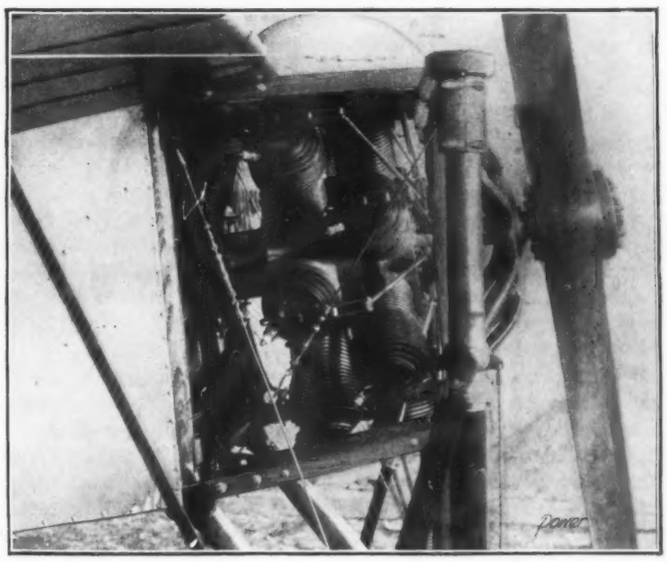


FIG. 8. SIDE VIEW OF 100-HORSEPOWER GNOME

The inlet-valve arrangement is unusual, even for a revolving-cylinder motor. It is located in the piston head, and is of the "automatic" type—that is, it is opened by the pressure of the air in the

crank case when the piston is pulled outward during the suction stroke. This arrangement makes it easy to deliver the working mixture to the engine; the carbureter delivery pipe merely discharges into the crank case through the stationary shaft, which is bored for that purpose, and the cylinders take their charges from the mixture in the crank case.

The exhaust valve of each cylinder is located in the head and opened by a radial push rod, as shown in Fig. 8. The push rods are moved successively by a cam on a short shaft inside the crank case, this shaft being turned by a gear which is driven at half speed by a gear mounted on the stationary shaft. No exhaust piping is used; the burned gases are discharged directly into the atmosphere from the outlets in the cylinder heads.

If the reader will first imagine the cylinders to be stationary and the crank shaft to revolve, the mechanical relations of the various parts will perhaps be clearer. Then remember that the cylinders and crank case revolve around the crank shaft instead of the shaft revolving within the crank case, and it will be obvious that the mechanical reactions and relative motions are exactly the same in the actual engine as they would be in the imaginary one.

A Puzzling Case of Premature Ignition

BY S. KIRLIN

In a certain power plant the load is carried on one unit, consisting of a tandem double-acting gas engine operating on natural gas. A great deal of difficulty was experienced at times due to premature ignition. The engine is rated at 300 horsepower; during the day time the load carried is only about 100 horsepower, but in the evening when the lighting load begins to come on the load rises to about the full capacity of the engine. As long as the engine was carrying the light load no trouble was apparent, but as soon as the load began to increase premature ignition would take place so frequently for about half an hour during the time that the heavier load was coming on that the engine would slow down to such an extent that the lights were very unsatisfactory; after about this length of time the trouble would gradually cease and would not occur again until the same time the next evening.

All of the valves and igniters were in perfect condition and the engineer was at his wits' end to find the cause of the trouble. One evening a short time before the load began to rise, some of the admission valves showed signs of sticking, and in order to clean out the deposit around the stems which was supposed to be the cause of the trouble, the engineer poured a small quantity of kerosene into the air inlet, allowing it to be drawn into the cylinders through the admission valves. The engine made several premature ignitions during the next few minutes but as the load was light they did not interfere with the speed, and the valves soon began working freely again. When the heavy load came on a few minutes later the engine gave no trouble, but picked up the full load and carried it through the run without any premature ignition whatever.

This gave the engineer a clue to the cause of the trouble, and each evening thereafter a small quantity of kerosene was poured into the air intake a few minutes before the heavier load was due to come on, and no more preignition trouble was ever experienced. The explanation seems to be that during the light-load period the temperature of the cylinders was not high enough to burn the deposits of carbonized oil which gradually collected on the heads of the cylinders and pistons, but when the heavier load came on the temperature rose to a point which ignited the deposits and produced premature explosions until all of the carbon was burned out of the cylinders. By admitting the kerosene, the carbon deposits were loosened and consumed in a few minutes instead of burning gradually as the load increased.

CORRESPONDENCE

Corrosion of Exhaust Pipe Drips

M. W. Utz in the November 8 issue tells of some troubles he had with water-cooled exhaust pipes and their drain pipes. I have had much the same experience with the exhaust-pipe drains of Diesel engines, and I think the exhaust gases are of similar composition in both cases. In one plant it was our custom to run as little water into the exhaust as possible, or just enough to keep the fuel valve casing cool, as the water from there was piped into the exhaust. Ordinary 1-inch black pipe would last about three weeks when running 24 hours per day and the nipples even a shorter period. I also tried galvanized pipe but the water could attack this where the threads were cut and, although it was longer-lived than black pipe, it did not solve the difficulty. I then tried extra-heavy iron pipe which endured for about three months. The exhaust pipes were of cast iron and gave no trouble.

Upon taking charge of another plant I found it had been the practice with the former engineers to run a much larger quantity of water into the exhausts than I had, which kept them much cooler and increased the life of the drain pipes far beyond anything I had experienced. In this plant there was one engine equipped with brass drains which ran for over a year without showing signs of failure.

My theory as to the cause of the corrosion is that the gases combine in a lesser or greater extent with the water to form a mixture similar to sulphuric acid, which contains two parts of hydrogen, four of oxygen and one of sulphur, as is shown by the formula, H_2SO_4. The one part of sulphur would be easy to obtain, as would the two parts of hydrogen, although there might be a scarcity of oxygen but not enough to make much difference. This acid would attack wrought iron very readily, but increasing the supply of water, since it is composed of two parts of hydrogen and one of oxygen, would change the nature of the combination and lessen the strength of the acid. This is further borne out by the action of the cooling water on any polished steel surface, which immediately becomes dull and blackened.

If I were in Mr. Utz's place I would have some of the water analyzed and find just what is formed. In any event, should the expense be permissible, I think lead-lined pipe such as is used for the conveying of acids would do for drain pipes and cast iron for exhaust piping.

G. H. KIMBALL.

East Dedham, Mass.

Cracked Flywheel Hubs

In the issue of POWER dated September 20, there was printed on page 1692 a letter by Frank E. Booth, entitled "Flywheels Broken by Preignition." It has been my experience that the breakage of flywheel hubs, in the location shown in the cut, is not caused by preignition but by improper methods in putting the flywheel on the shaft. The taper key is driven into the keyway when the bolts in the hub are loose, causing one-half of the flywheel hub to ride on the key instead of on the shaft. The bolts are then pulled down, throwing a tremendous strain on that section of the hub which Mr. Booth's drawing shows broken. This method of securing the flywheel will almost invariably crack the hub. The hub may not necessarily break at the time the bolts are pulled down, but the strain can be thrown into the hub of the wheel so that this breakage occurs at some later time. The proper method is, of course, to pull down the bolts perfectly snug on the shaft before the key is driven home.

A. Y. EDWARDS,

Chief mechanical engineer, Foos Gas Engine Company.

Springfield, O.

Quite recently, within the state of Pernambuco, Brazil, was discovered what promises to be a most valuable coalfield. The area embraces about 22 square leagues.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Power Plant Design and the Operating Engineer

Every day one sees mistakes in the layout of power plants, caused by the designer being thick headed, or perchance it may have been a consulting engineer or the chief engineer in charge of operation or it may have been the fault of the superintendent or the ideas of the president of the company.

I believe, emphatically, that in the majority of cases blunders in power-plant design are due more to self-inflation than any other cause.

Operating engineers frequently say, "The engineer in charge is best qualified to pass on contemplated improvements"; again, the consulting engineers laugh up their sleeves over mistakes made by operating engineers, which mistakes were due solely to ignorance of the laws of philosophy, simple laws which every man should know thoroughly before attempting to design or be a critic in laying out a power plant. How can the engineer in charge secure uninterrupted and satisfactory service while watching the hundred and one things around a plant, continually looking where improvements can be made and at the same time be a designing engineer? Is not the old adage, "He is a jack of all trades and master of none," quite true? This is a day of specialists, and any man who desires to aspire to any degree of success must specialize along one particular line of work.

I do not wish to infer that the operating engineer should not be consulted when changes are to be made, but that others who have made power-plant designing a study should be consulted, thus securing the cardinal ideas of both. The consulting engineer is not expected to come in offhand and operate a power plant with any degree of satisfaction; then why should the operating engineer expect to be eligible for designing? To be a success along any particular line a person must devote his efforts without reserve along that line.

The secret of success in any concern at the present time is organization and system; this can be obtained only by securing men of ability for each individual line of work, such that every phase of the organization may secure faultless results. This factor of strong organization and system is responsible in a greater degree than any other thing for the monstrous success of the central stations forcing so many isolated plants to close down. The only course left for the isolated plants to pursue in order that they may withstand the onslaught of the central station is to be under the supervision of an able consulting engineer, thoroughly uptodate in operation and design, but not just an operating engineer, by any means.

If the operating engineer would make himself eligible as a designer, he must aspire to a higher plane of education and social standing, such as is gained through constant study and observance, to which should be added the constant association with men of brains and achievements.

WALDO WEAVER.

Middletown, O.

Gas in the Pump

I had a duplex pump which was used to pump water from the river through a lift of about 15 feet, and which gave entire satisfaction in this capacity. However, as the river water was very poor, we had two wells driven and connected the suction of the pump to these wells, with no misgivings, as the lift was only about 8 feet.

The pump worked all right for a few minutes and then started to buck, as if it were drawing air. I measured the hight of the water in the wells, and finding the lift to be less than 15 feet, I took a candle and carefully looked over the suction line, trying every joint, but did not succeed in locating a leak. I then looked over the valves and packed the water end, but everything seemed to be in good order; still it continued to buck when run at normal speed, which was considerably below 100 feet per minute.

The pump was handling water all the time, but not enough to supply us, and every attempt to make it do so resulted in failure.

That evening on passing one of the filter tanks, I noticed that the pressure was above normal, and proceeded to cut it out and flush the mud into the sewer. I had just started, when there was a slight explosion, filling the cellar with flames. In less than a minute it was all over and I went back, although there was still a flame about two feet long coming from the flush pipe of the filter. This was in an oil section, and the water wells were in the vicinity of many old oil wells, which accounted for the trouble with the pump and the explosion. It developed that the pump had been handling water and natural gas mixed, the filters had formed a pocket for the gas, and when I liberated it in flushing them, a gas jet on the wall ignited it, and the explosion occurred.

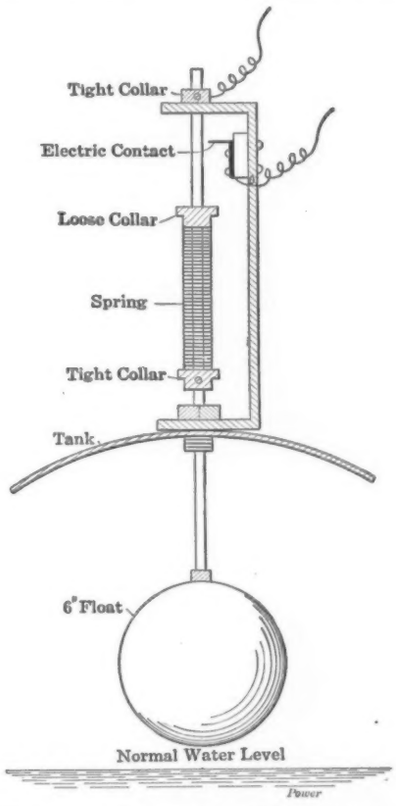
I put a pet cock on each one of the filters, and lighted that part of the cellar from them, but I am still unable to pump the water through a lift greater than 15 feet.

A. STEVENS.

Renfrew, Penn.

High Water Alarm

In the boiler room of a large industrial plant were located several vented return tanks into which the condensation resulting from the manufacturing processes and from the heating system was returned. Due to the unsteadiness with which the water returned and the fact that the pump room was not under constant supervision, the tanks occasionally filled up, the water backing into the return system with serious results. Therefore, a ¾-inch hole was drilled and tapped in the top of one of the tanks and a brass bushing with a hexagon head and



FLOAT AND ELECTRICAL CONNECTIONS

a 5/16-inch hole was fitted to the tank. A piece of ⅜x1½-inch flat iron 18 inches long was bent, as shown, and at one end was drilled a ¾-inch hole, by means of which it was secured to the tank under the head of the bushing, and at the other end was drilled a 5/16-inch hole alining with the one in the bushing. A 6-inch float attached to a 5/16-inch brass rod was provided and held in place by a small collar at its upper end, this being adjusted so that when the water was at its normal level the float was above it. A brass collar with a wide flange was screwed to the lower part of the rod and a similar collar was fitted to slide freely on the rod with a light coiled-brass spring between the collars. At the side of the frame was fastened a pair of contacts which were attached in such a position as to close a bell circuit when the sliding collar came in contact with them. As the level of the water rises the float lifts the loose collar against the contacts and rings the bell; further upward movement of the water compresses the spring and leaves the bell in operation. The apparatus as described has answered the purpose admirably and can be made with material which can be readily picked up around most any plant.

LEWIS C. REYNOLDS.

Willard, N. Y.

A Convenient Flue Cutter

The accompanying illustrations show a device for cutting out flues from a locomotive type of boiler. The tool holder is first inserted in the flue, the cutting tool is placed in the holder and the point is driven through the tube. The socket *A* is then attached to the tool holder at *A*, the 1⅛-inch steel rod with a square on one end to fit the spindle *Y*, and an eye on the other end which is welded into socket *A* at *B*. This rod drives the cutter to be inserted into any flue without changing the device. One revolution cuts off the flue.

Fig. 1 shows the assembled device and the method of hanging it up with a block and fall. The two chains shown on the lower part of the device are very important. These are anchored to each frame to steady the device and to keep it from swinging around when cutting off a flue. The large gear wheel is mounted on the spindle *Y*, which drives the cutter, and the power of the motor is transmitted through the gears *V* and *W*. The ease with which this device can be adjusted from one flue to another is a feature that should recommend it to anyone.

W. H. SNYDER.

Stroudsburg, Penn.

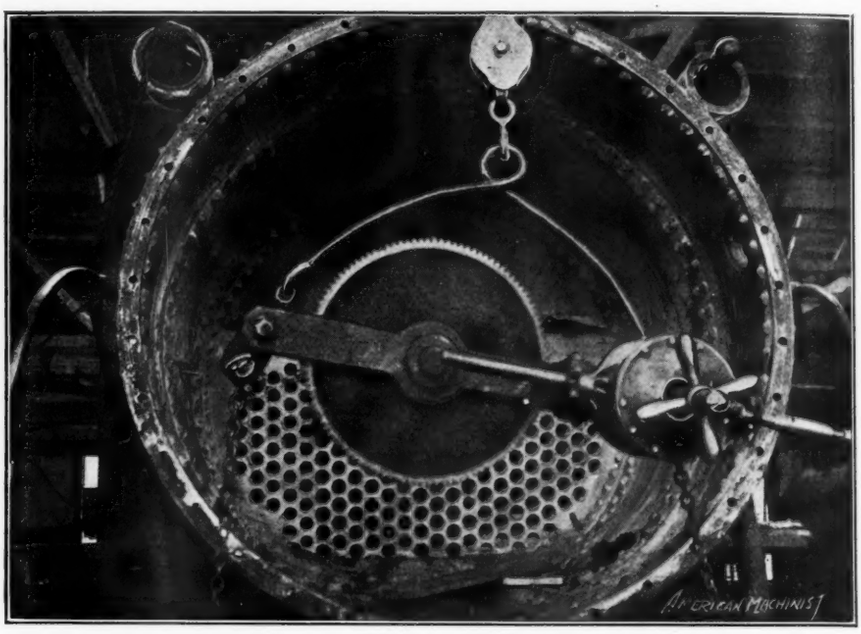


FIG. 1. FLUE CUTTER IN POSITION

Color Scheme for Piping

Any color scheme for general adoption must be simple in form and broad in scope. Both of these essentials are entirely neglected by the majority of proposals for color schemes. The trouble lies in the consideration of details to the neglect of fundamentals, resulting in schemes not only devoid of method, but much too cumbersome for general utility.

A standard color scheme for piping must be capable of consistent application to any and every plant. Plants are not

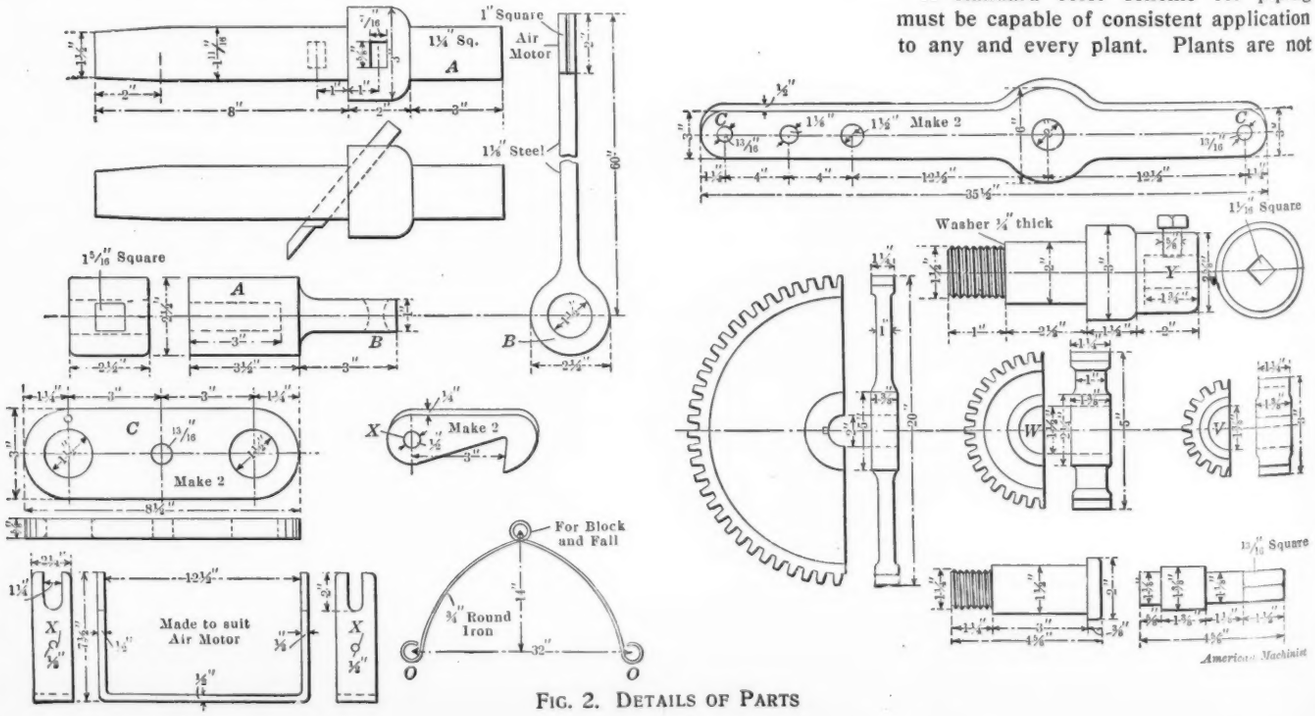


FIG. 2. DETAILS OF PARTS

all steam nor power plants and by no means should industrial plants be neglected. Industrial plants are extensive and complicated and in far greater need of some system of pipe identification than are the majority of power plants. The list of materials conveyed by piping in the various industrial plants would exhaust any possible color scheme, unaided by Roman letters and Arabic numerals. Therefore, little thought should convince one that in such an extensive system, colors can only aid and not supplant language.

Materials conveyed through pipes are in one of two physical conditions: either liquid or gaseous. Solids are occasionally conveyed but they are usually held in suspension by either a gas or a liquid. The liquid or gaseous state is either stable or unstable within the range of the temperature and pressure employed at the plant. The terms stable and unstable are used here not in an absolute sense but in a relative sense, the condition within the pipe to that in the atmosphere, subject to the change that would occur at the bursting of the pipe or opening of a valve. Even a gas is not absolutely stable, though within ordinary working pressure and temperature it is decidedly so. A vapor, though not a gas, is in a gaseous state and is unstable. A liquid that would remain a liquid at the bursting of a pipe would here be considered a stable liquid, while a liquid that would flash into vapor at the release of pressure would be unstable. This gives the four principal conditions of materials carried by piping and suggests a color scheme of the following character:

Gas	Black
Vapor	White
Stable liquid	Olive
Unstable liquid	Maroon

Some method of distinguishing relatively high and low pressures can be added by representing generator outlets and motor inlets by plain-colored valves and fittings, and by coloring yellow the valves and fittings of generator inlets and motor outlets. The term generator is used here to designate any apparatus doing work upon the fluid and boosting the pressure, and the term motor as any apparatus operated by the fluid and through which the pressure drops.

This color outline is adaptable to any plant. Any addition would but limit its scope. The further working out of details should be left to the managements of individual plants and this can be carried out either with colors or preferably with stencils. The stencil requires no more maintenance than does the plain color and is certainly simpler and more comprehensive. A stencil can show the material, name, pressure, temperature and direction of flow if advisable. It is frequently found necessary to post directions and cautions at important valves.

On lines that are difficult or impossible to trace, some identification should be permanently stamped on the valve body or upon a metal tag secured to the valve or line.

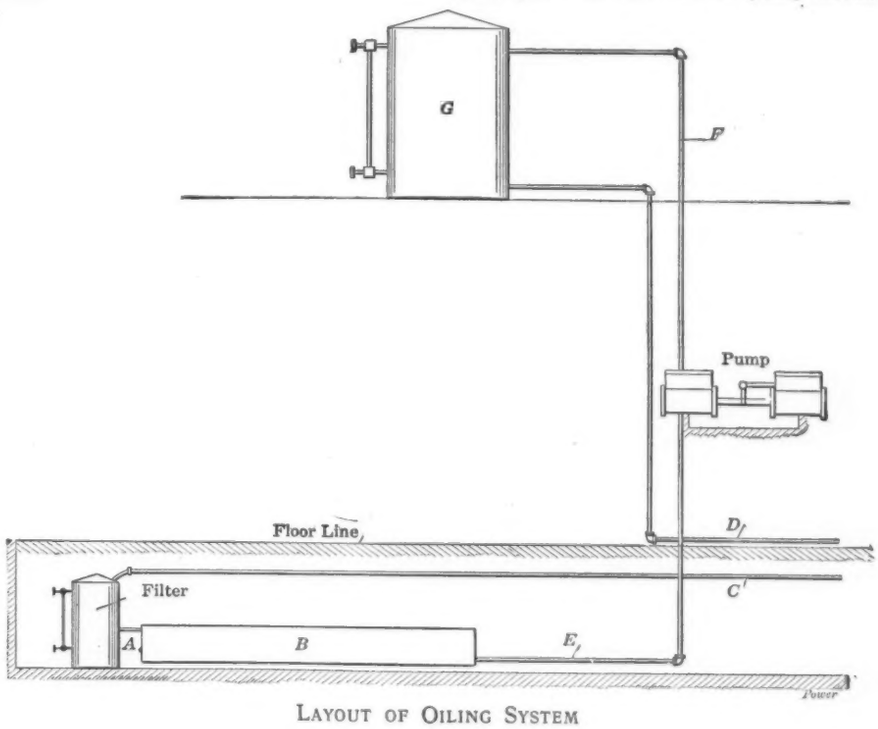
In addition to the foregoing methods of identification there should be an accurate diagram of the piping layout. However, with most industrial plants, especially with the older ones, the drafting of such a diagram would be next to impossible. The necessity of an accurate and convenient identification of piping cannot be overestimated and the standardizing of this identification as far as possible would be welcome.

J. W. TAYLOR.

Massillon, O.

A Homemade Oiling System

I have never been able to understand why some engineers will slop around day after day filling oil cups by means of a squirt can, when in most plants an automatic oiling system can be installed with little or no expense. In most plants there is enough scrap of various kinds lying around with which to install the system, and there is usually some man on the job who can do the work during his spare time.



LAYOUT OF OILING SYSTEM

The illustration shows the system which was installed in a plant of which the writer had charge. There were two 14x14-inch high-speed engines, each equipped with a large oil pot, the oil being piped to the various bearings from this pot. After lubricating the bearings the oil ran down into the base of the engine, from which it had to be drained out into a bucket, poured into the filter, drained through the filter into another bucket, and again poured into the oil pot. This arrangement not only required considerable time and careful watching, but was sloppy, wasteful and unsightly. With the new arrangement, however, the oil instead of being caught in a bucket was led through pipe *C* to the filter, which was placed in a pit under the engine-room floor; after being filtered, it drained through the pipe *A* into the storage tank *B*. It was then pumped through the pipes *E* and *F* to the storage tank *G*, from which it was led back by pipe *D* to the oil pots on the engines. These oil pots were fitted with tight covers so that the pressure due to the head of tank *G* was on the needle valves, which regulated the feed to the bearings. A stream of oil was kept running over all the bearings and much better lubrication was secured with less trouble and cost. It was necessary only to run the pump about five minutes to fill the tank *G*, and after being filled it lasted from six to eight hours, the oil running over the bearings and returning through the filter into the storage tank *B*.

Nearly all the material used was taken from the scrap pile, tank *G* and a few pipe fittings being all that had to be purchased. Tank *B* was made from a piece of ventilating duct, picked from the scrap pile. If there is no pump about the plant that can be used for the purpose, a small plunger pump can be made easily from pipe fittings and be attached to the end of the engine shaft, rocker arm or valve stem. Even though the whole apparatus had to be purchased, it would soon pay for itself in the saving of oil. The first month the system was in operation, only seven gallons of oil were used on both engines.

R. L. RAYBURN.

Kansas City, Mo.

Questions Before the House

The Theory of the Cooling Tower

Joseph H. Hart's "Theory of the Cooling Tower" as expounded in the October 4 issue contains some statements which, if true, are of great importance to all owners and operators of cooling towers. The most important of these statements and at the same time the one which I believe is based more upon theory than upon actual tests is that in which he recommends the warming of the air before its entrance into the cooling tower. His claim is that by first passing it through the boiler or engine-room loft before blowing it into the cooling tower the evaporation is increased and the efficiency of the tower is improved. He reasons that since more water is evaporated, more heat becomes latent and the water is cooled to a lower temperature. Here is where I believe he is in error. What really occurs when the temperature of the air is raised, is that the heat which becomes latent owing to the increased evaporation comes largely from the air, and the final result is that the water will not be cooled to as low a temperature as it would be, had the air been left in its natural state.

That this is true can probably best be shown by an example. Assume the temperature of the air to be 60 degrees, the barometer 30 inches and the psychrometer or hygrometer readings to be, dry bulb, 60 degrees, and wet bulb, 55 degrees. According to the psychrometric tables as published and used by the United States Weather Bureau, the temperature of the dew point would then be 51 degrees, and the absolute humidity, 4.222 grains of water vapor per cubic foot. Now, suppose that this air were warmed in some way to a temperature of 96 degrees. According to the experiments of Regnault, Herwig and others, the coefficient of expansion of a mixture of air and water vapor is very nearly the same as for dry air. Then, one cubic foot of the air at 60 degrees would expand to $1 + [(96 - 60) \times 0.00203252] = 1.073$ cubic feet at 96 degrees and, each cubic foot at 96 degrees would contain

$$\frac{4.222}{1.073} = 3.935 \textit{ grains}$$

of water vapor. According to the psychrometric tables, 3.935 grains per cubic foot corresponds to the dew point of 49 degrees, nearly, and when the air temperature is 96 degrees and the dew point 49 degrees, the difference in temperature between the dry- and the wet-bulb thermometers is 28.7 degrees, or the

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

wet-bulb thermometer shows a temperature of

$$96 - 28.7 = 67.3 \textit{ degrees.}$$

This is the temperature of evaporation and the lowest temperature to which water can be cooled by passing air over it whose temperature is 96 degrees, and whose humidity is 3.935 grains of water vapor per cubic foot. But, the temperature of the wet bulb or the temperature of evaporation was 55 degrees before the air was warmed. So, it will be seen that by raising the temperature of the air from 60 to 96 degrees, the moisture per pound of air remaining the same, the temperature of evaporation is raised from 55 to 67.3 degrees or the possible cooling effect lessened by

$$67.3 - 55 = 12.3 \textit{ degrees.}$$

Of course, in an actual cooling tower the water is never cooled to as low a point as the temperature of evaporation.

Now, by means of a somewhat lengthy calculation it can be shown that this loss in cooling effect on the water, in spite of the increased evaporation, is caused by the fact that the air must also be cooled. Mr. Hart seems to think that he can avoid this cooling of the air by speeding up the fan and by shortening the path of the air through the tower. It is true that this may not give the air time to cool, but neither will there be time for it to become saturated and, if the heating effect by conduction of the air on the water is lessened, the cooling effect due to evaporation is also lessened, and to the same extent. This can be shown to be true by a very simple experiment. Take a sling psychrometer and whirl it with just sufficient rapidity to obtain an accurate reading and then whirl it very rapidly. It will be noticed that the temperature of the wet bulb is not materially changed. In fact, owing to the increased friction of the air it may show a slight rise. The only thing to be gained by increasing the velocity of the air through the tower is a nearer approach of the temperature of the water leaving the tower to the temperature of evaporation, though the former will never equal the latter as long as the water is flowing.

Mr. Hart also makes the statement that after the air has risen the first 2 to 4 feet in the tower and has become saturated, it then does its cooling solely by conduction, convection and radiation. This is not strictly true because, as the air rises in the tower, it comes in contact with warmer and warmer water, cools the water by conduction and itself becomes warmer; the vapor it carries is then no longer saturated and more water is evaporated. In fact, in the upper part of the tower the cooling effect due to evaporation greatly outweighs that due to conduction.

Mr. Hart uses the unscientific expression "the absorptive power of air for water vapor." If air has an absorptive power for water vapor it is so small that no experimenter has been able to determine it. When water exposed to the air evaporates, it does not evaporate because of the air, but in spite of it. Air is always a hindrance to evaporation because the water vapor must diffuse through the air as fast as it is formed and unless the air is in motion this is a slow process.

A. E. MUELLER.

Mayville, Wis.

Right Angled Triangular Weir

In the article on "The Right Angled Triangular Weir," by Charles N. Cross, which appeared in the September 20 number, the author, in framing a formula for weirs of this kind, assumes

$$h = \textit{the head of water.}$$

The illustration and description show this to be his assumed head of water above the lowest part of the notch. And, having a right-angled triangular weir, h^2 = the area of the notch filled up to the level of the assumed greatest head acting on the weir, and equal to the area of cross-section of the theoretical discharge.

With these conditions he concludes that because the theoretical velocity at the lowest part of the notch would be $1\sqrt{2gh}$, "therefore, the theoretical discharge is,

$$q = h^2 1\sqrt{2gh}"$$

i.e., he concludes that "the theoretical discharge" of a weir is to be taken as equal to the assumed cross-sectional area of the discharge, multiplied by the theoretical velocity due to the greatest head acting on any part of the cross-sectional area.

Theoretical discharge, like actual discharge, is equal to the product obtained

from multiplying the assumed cross-sectional area by the assumed mean velocity of water traversing the cross-section. Hence, in order for the formula to be consistent for representing the theoretical discharge of a weir wherein h^2 represents the cross-sectional area, the other factor, $\sqrt{2gh}$, would have to represent the mean velocity. But, when h is the total head, the assumption of $\sqrt{2gh}$ as the value of the factor representing the mean theoretical velocity is

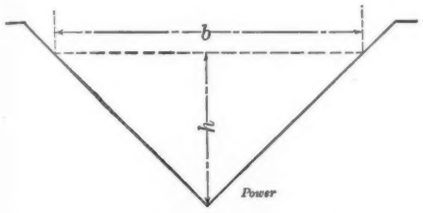


FIG. 1. RIGHT-ANGLED TRIANGULAR WEIR

erroneous, whether used with reference to triangular or any other form of weir, because the heads which create velocity at different parts of the cross-section have values which vary between h and zero. The velocity at different parts of the cross-sectional area must, therefore, vary between $\sqrt{2gh}$ and zero, and the mean velocity can be only a fractional part of $\sqrt{2gh}$.

The theoretic discharge of a notch, whether triangular, semi-circular, or other form of open-sided figure, may be assumed as made up of a number of narrow rectangular notches of uniform width placed side by side and filling the cross-sectional area under consideration. Referring to Fig. 1, which is like the right-angled triangular weir illustrated in the article of Mr. Cross, wherein h is the depth and

$$b = 2h$$

is the breadth of the notch filled, the

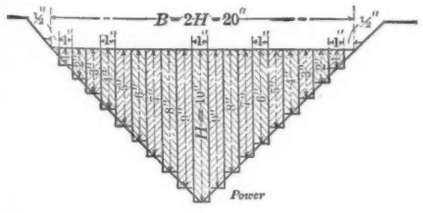


FIG. 2. DIAGRAM OF WEIR DIVIDED INTO A NUMBER OF NARROW RECTANGLES

cross-sectional area may be supposed to be composed of narrow rectangular weirs of uniform width placed side by side as illustrated in Fig. 2.

The discrepancy between the total cross-sectional areas due to stepping downward the narrow weirs to different depths need be very little and resulting discrepancies due to head may be almost entirely eliminated, though the width of each section is taken as a considerable proportionate part of the width of the notch. For a right-angled triangular weir, if the width of sections is taken as equal to 1/20 of the total width b, Fig. 1, as indicated in Fig. 2, then the summation of discharges of the different sections will be found to be within a small fraction of 1 per cent. of the discharge of the notch computed by the refined methods of calculus.

The theoretical discharge of a rectangular notch or weir wherein

L = Width in feet, and
H = Total head in feet

is given in cubic feet per second by the formula:

$$Q = L H^{\frac{3}{2}} \sqrt{2gh} \qquad (1)^*$$

In Fig. 2, assuming the width of each section to be $L = 1$, the total depth of the notch as 10 and the total width as 20, then we would have a central section in which $L = 1$ and $H = 10$, and nine weirs each side of the central section having $L = 1$ and values of H from 9 to 1, and two small triangles at the extreme right and left sides which may be neglected.

The sum total of discharges of these weir sections, computed from equation (1), will be found to equal 1354.79 + cubic feet per second. Computed by the equation

$$q = h^2 \sqrt{2gh}$$

in which h, for a right-angled triangular weir, is taken to be 10, we would obtain a theoretical discharge of 2532.76 + cubic feet per second, from which we see that the quantity obtained by the summation of rectangular weirs is about 53.45 per cent. of the quantity computed by the last named equation, which is the formula for the theoretical discharge given in the article referred to.

A similar result is obtainable by dividing the triangular weir into horizontal sections and taking the sum of their discharges, considering each section to discharge with a velocity due to the head above it. A precise formula for the theoretic discharge of a triangular weir which is founded on the summation of discharges of horizontal sections reduced by calculus gives

$$Q = \tfrac{8}{15} B H \sqrt{2gH} \qquad (2)$$

wherein

Q = Discharge in cubic feet per second;
B = Breadth of notch in feet;
H = Total head (or "depth of notch filled") in feet, and
g = Acceleration due to gravity (= 32.16).

For a rectangular triangular weir, in which $B = 2H$, the formula becomes

$$Q = \tfrac{8}{15} \times 2H \times H \sqrt{2gH} = \tfrac{8}{15} H^2 \sqrt{2gH} \qquad (3)$$

or representing the coefficient for actual discharge by K we have the usual formula for the actual discharge of a 90-degree notch, viz.,

$$Q = \tfrac{8}{15} K H^2 \sqrt{2gH} \qquad (3a)$$

or its equivalent,

$$Q = \tfrac{8}{15} K H^{\frac{5}{2}} \sqrt{2g}$$

It is to be observed that the fraction 8/15 in the second member of (3) and in the formulas following indicates that the theoretical discharge = 53⅓ per cent. of $H^2\sqrt{2gH}$. The discharge obtained by computing vertical sections as illustrated by Fig. 2 being 53.45 per cent. of the discharge computed as $q = h^2\sqrt{2gh}$, H in one case being the same as h in the other, it is of interest to note that the result obtained by dividing the notch into narrow rectangular weirs, and taking the sum of their theoretic discharges, gives a discharge which is within 0.03 of one per cent. of the theoretic discharge obtained by the employment of formula (3), which is based upon the most refined methods of mathematical analysis.

FRANKLIN VAN WINKLE.

Paterson, N. J.

*For demonstration see POWER AND THE ENGINEER, October 5, 1909, page 567.

Indicator Reducing Motion

Having had some experience with the indicator reducing motion described in the issue of October 11, I write to point out some of the disadvantages of it which led to its being replaced by the well known reciprocating lever type.

The attractive features of the reciprocating wheel type of reducing motion are its neatness, its correctness and the fact that it can be set in motion at any time, there being no need to stop the engine to attach the gear to the crosshead.

It was proposed to adopt the gear as the standard for a line of horizontal engines which would be run at any speed up to 180 revolutions per minute. It was recognized from the first that there might perhaps be some risk of trouble due to the effect of the inertia of the wheels in prolonging the travel of the indicator drum when the crosshead came to the end of its stroke, and this was guarded against by having aluminum wheels as light as possible.

When this gear was tried, it was found that with an engine running at about 120 revolutions per minute, there were unmistakable signs of this inertia effect shown on the indicator diagrams, and at speeds above 120 revolutions per minute the gear was quite unreliable from this cause.

It was seen that when moving in one direction the indicator-drum spring added its pull to the inertia of the wheel and that when moving in the other direction the drum spring acted against the inertia of the wheel. Various refinements were tried to make this gear satisfactory, but they were unsuccessful and the gear was abandoned in favor of the reciprocating-lever type, which has the advantage of being cheaper.

In the reducing motion supplied by indicator makers and attached to the bottom of the indicator, there is no inertia effect of the wheels to affect the length of the diagram, as during the outward stroke of the crosshead the indicator cord winds up a spring and this spring overcomes the inertia of the wheels, while on the inward stroke of the crosshead the cord is held taut by the tension of the spring and by the nonstretching of the cord there is no overrun of the indicator drum when the crosshead reaches the end of its stroke.

JAMES CANNELL.

Stanford le Hope, Eng.

Leakage through a Piston Valve

In the October 11 issue appeared an article under the above title. That leakage occurs past any valve, whether piston, flat, Corliss or other type, is, of course, recognized, and for years the efforts of designers and builders have been constantly put forth to reduce this loss to a minimum, while at the same time considering the other factors that enter into the building of a successful engine.

To measure accurately the amount of leakage past the valve of an engine which is in motion is a difficult problem, and it is doubtful if the results as determined by Mr. Mitchell's laboratory method can be relied upon as properly representing actual performance under normal conditions. Even assuming the method employed in the tests reported as representing actual working conditions, as nearly as such can be approximated, what reason is there for assuming that similar results would not be obtained if this same method were applied to a flat valve, by making one end of the valve wide enough to lap the port, entirely preventing its opening, or to a Corliss valve by special construction?

This method of testing the valve by allowing it to move over an unported surface and measuring the water escaping by the "blind end," is in effect duplicating the condition which exists in certain forms of single-acting engines. The steam is on one side of the piston only, and with the constant changing of temperature, due to the movement of the piston, there is of necessity leakage of water of condensation no matter how well the piston is fitted. Much depends, of course, upon the fit of any valve, and the leakage test reported by Mr. Mitchell is very incomplete, in that it does not report the total steam consumption of the engine, for this would be a guide as to its condition, which the facts presented do not directly yield.

It is possible, of course, from the curve in Fig. 6, of Mr. Mitchell's article, to draw some conclusions, but results so obtained are so ridiculous as to create uncertainty that the curve is understood.

If, for example, this curve represents leakage per hour, then at 15 horsepower, the normal load of the engine, there was approximately 200 pounds leakage, or 13⅓ pounds per horsepower per hour.

This leakage is represented as being 22 per cent., approximately, of the total steam consumption, which would mean that the steam consumption of the engine was about 60 pounds per horsepower per hour. This is nearly twice what an engine of this type, even of so small a size, would consume if in reasonably good condition.

There are on record many authentic tests of engines with solid piston valves, which show results considerably better than 30 pounds, and it is safe to assert that these results have not been surpassed by any other form of single-valve engine.

From another viewpoint, if it should be accepted that the conclusions drawn by Mr. Mitchell from his tests, that 22 per cent. of the total steam consumption of the engine is attributed to valve leakage, and that this amount is to be looked for in all engines of this type, then it would be proper to assume that the results of recorded tests could be bettered by that amount if some other form of valve had been used.

To demonstrate the absurdity of such an assumption, there follows a brief summary of two such records. These tests are selected as fairly representative of the type of engine under discussion, and because the engines were of small size, although not so small as that upon which Mr. Mitchell's tests were conducted.

The first test was conducted by Professor Diederichs, of Cornell University. The engine had been in operation prior to the test for four years, running much of the time day and night.

Type of engine, single-cylinder automatic with solid piston valve.	
Size of engine, inches	13x12
Revolutions per minute (average)	262.17
Steam pressure (average), pounds	118.3
Quality of steam, per cent	98.65
Indicated horsepower	106.7
Steam per i.n.p. per hour, pounds	27.05

The second was a test made by the engineers of the War Department, United States of America. It was an acceptance test, the engine being new.

Type of engine, single-cylinder automatic with solid piston valve.	
Size of engine, inches	9½x11
Revolutions per minute (average)	304.2
Steam pressure (average), pounds	102.5
Quality of steam, per cent	97.54
Indicated horsepower	56.58
Steam per i. h. p. per hour, pounds	28.63

This second test, it will be observed, was of an engine considerably smaller than the first, and the steam pressure was lower, which factors probably account for the difference of approximately 1½ pounds in the results obtained. Both engines were manufactured by the Harrisburg Foundry and Machine Works.

That the results given could not have been bettered to the extent of 22 per cent. is evident, for this would mean 21 pounds in one case and 22⅓ pounds in the other, which are results never approached with single-cylinder engines equipped with any type of single valve using low-pressure saturated steam and operating noncondensing.

It is possible that there may be some advantage in the use of rings in the valves of larger engines, where the relative cost of renewal is great, but the test by Professor Diederichs given above is evidence that the wear of a piston valve cannot be very great if such results as obtained in this case can be had after four years of hard service. If rings are used, they should be of the self-adjusting type.

Outside the question of economy, the piston valve has many notable advantages over other forms of single valves and, as far as the economy is concerned, the results have not been surpassed. Its very general and growing use in marine and locomotive practice is strong evidence of the favor in which it is held by leading engineers of this and other countries and not without reason.

The piston valve is the lightest form of valve and can readily be perfectly balanced, thus reducing wear to a minimum and causing the smallest amount of friction and strain on the governing mechanism.

Changes of temperature do not distort it, as is the case with square or rectangular forms of valves, working between the seat and a heavy pressure plate.

High steam pressure and superheat can be safely used.

The valve-chest covers and stuffing boxes are only subject to the pressure of the exhaust.

Greater port openings are afforded than with any other form of single valve, resulting in less wire drawing and better steam distribution.

Repairs and replacement can always be quickly and cheaply effected.

These and other arguments can be summed up in the statement that the piston valve is the most perfectly balanced, quietest running and simplest form of single valve, and whatever the leakage may be past a properly fitted valve of this type, it is certainly not any greater, if as great, as past other forms of single valves, as is attested by the comparative results obtained from engines using this and other forms.

Investigations such as those made by Mr. Mitchell are always interesting, but in reporting them there should be given the full data of the tests, and care should be exercised in drawing conclusions, for otherwise injustice may be unwittingly done and progress rather retarded than advanced.

BENJAMIN T. ALLEN,

Chief Engineer, Harrisburg Foundry and Machine Works.

Harrisburg, Penn.

One Valve Performing the Functions of Two

Under the above caption in the issue of October 18, B. U. Potter asks for opinions as to the advisability of the application of his device to a compound pump.

If the high- and low-pressure cylinders could be placed side by side close together, his arrangement would probably work out good, but with the usual arrangement of a compound pump, that is, with the high- and low-pressure cylinders placed tandem the scheme involves an increased steam consumption owing to the volume of the pipe connecting the cylinders. The following explanation based on assumed volumes for cylinders and connections shows the comparative weight of steam used for the usual construction, and for that proposed by Mr. Potter.

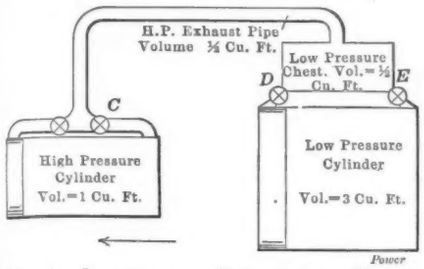


FIG. 1. DIAGRAM OF PUMP WITH USUAL VALVE ARRANGEMENT

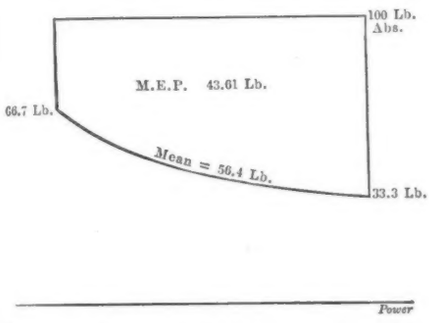


FIG. 2. INDICATOR DIAGRAM FOR HIGH-PRESSURE CYLINDER

The usual construction is diagrammatically represented by Fig. 1. Assume steam at 85 pounds gage pressure. The pistons having just completed the stroke in the direction of the arrow, the high-pressure cylinder is full of steam at 100 pounds absolute pressure; the high-pressure exhaust pipe, low-pressure steam chest and low-pressure cylinder are full of steam at terminal pressure due to expansion. Since the ratio of the cylinders is 3 to 1, this terminal pressure should be

$$\frac{100}{3} = 33.3 \text{ pounds}$$

absolute. The next thing that happens is the shifting of all of the valves, E closing and C and D opening. At this time the pressure in the high-pressure cylinder, the high-pressure exhaust pipe and the low-pressure chest equalize and one volume at 100 pounds plus one volume at 33.3 pounds results in two volumes at 66.7 pounds. Then, the pistons travel in the opposite direction and expansion occurs. At the end of expansion, the exhaust pipe, chest and low-pressure cylinder are full, this being equivalent to four volumes, thus the two volumes at 66.7 pounds have expanded to four volumes at 33.3 pounds. The theoretical diagrams from the cylinders are shown in Figs. 2 and 3.

For two expansions the ratio of mean to initial pressure is 0.846; therefore, the mean pressure of the high-pressure exhaust and the low-pressure admission is

$$66.7 \times 0.846 = 56.4 \text{ pounds.}$$

The mean effective pressure in the high-pressure cylinder is

$$100 - 56.4 = 43.6 \text{ pounds.}$$

The mean effective pressure in the low-pressure cylinder is

$$56.4 - 15 = 41.4 \text{ pounds.}$$

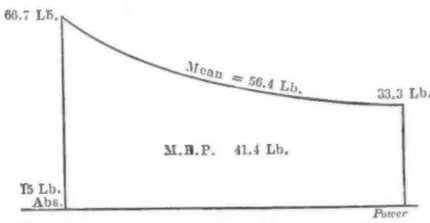


FIG. 3. INDICATOR DIAGRAM FOR LOW-PRESSURE CYLINDER

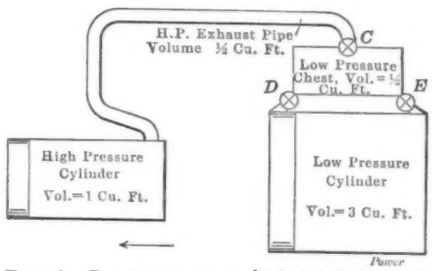


FIG. 4. DIAGRAMMATIC ARRANGEMENT OF PUMP WITH SINGLE VALVE

The areas of the cylinders are proportional to their volumes. Assume the area of the high-pressure cylinder to be 100 square inches and that of the low-pressure cylinder, 300 square inches. Then, the total pressure is found as follows:

$$100 \times 43.6 = 4{,}360 \text{ pounds}$$
$$300 \times 41.4 = 12{,}420 \text{ pounds}$$
$$16{,}780 \text{ pounds}$$

total pressure on the pistons. At each stroke, the high-pressure cylinder takes in 1 cubic foot of steam, which at 100 pounds absolute pressure weighs 0.23 pound. Therefore, each pound of steam represents a total combined pressure on the pistons of

$$\frac{16{,}780}{0.23} = 73{,}000 \text{ pounds.}$$

Now, making the same assumptions as to volumes and pressures, the Potter proposed construction is indicated diagrammatically in Fig. 4.

On the opening of the valves the high-pressure cylinder and the pipe are full of steam at initial pressure, hence there are 1½ volumes at 100 pounds absolute pressure. Also, the low-pressure chest contains steam at the terminal pressure of the low-pressure cylinder. From this, the high-pressure diagram works out as per Fig. 5, and the low-pressure as per Fig. 6.

The mean effective pressure in the high-pressure cylinder is

$$100 - 72.7 = 27.3 \text{ pounds.}$$

The mean effective pressure in the low-pressure cylinder is

$$72.7 - 15 = 57.7 \text{ pounds.}$$

Then, the total pressure is:

$$100 \times 27.3 = 2{,}730 \text{ pounds}$$
$$300 \times 57.7 = 17{,}310 \text{ pounds}$$
$$20{,}040 \text{ pounds}$$

This total pressure is larger than in the previous case but in order to secure it there has been required both the high-pressure cylinder and the pipe full of live steam or 1.5 cubic feet at 0.23 pound per cubic foot or 0.345 pound of steam.

Therefore, each pound of steam represents a total pressure on the pistons of

$$\frac{20{,}040}{0.345} = 58{,}072 \text{ pounds.}$$

Thus, under the conditions assumed above, a pound of steam will do about 12.5 per cent. more work with the regular construction than with the construction proposed by Mr. Potter. The principal idea in compounding a steam pump is to save steam and therefore the proposed scheme is a step in the wrong direction and any mechanical advantages are more than offset by the loss of economy in operation.

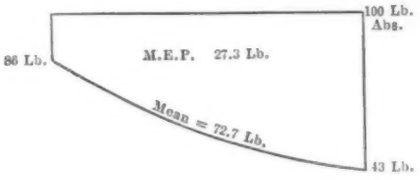


FIG. 5. HIGH-PRESSURE DIAGRAM FROM SINGLE-VALVE PUMP

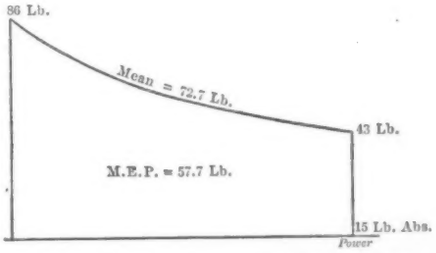


FIG. 6. LOW-PRESSURE DIAGRAM FROM SINGLE-VALVE PUMP

C. H. MAYO.

South Hadley, Mass.

Firing with Natural Gas

In the October 18 issue, Mr. Wilson asks, among other things, the amount of natural gas required per horsepower for return-tubular boilers, and the amount of water that 100 cubic feet of gas will evaporate from and at 212 degrees Fahrenheit. It would be difficult to give a specific answer with the limited amount of information given, but it is quite easy to arrive at an approximate answer.

The amount of heat contained in natural gas is a rather variable quantity, but 1000 B.t.u. per cubic foot would probably represent a fair average. Assuming gas of this quality to be used and that the efficiency of the boilers is 75 per cent., we can readily compute the amount of gas required per horsepower. The accepted equivalent of a boiler horsepower is 33,305 B.t.u. To deliver this amount of heat to the boiler

$$\frac{33{,}305}{0.75} = 44{,}406\ B.t.u.$$

will be required. This amount of heat will be supplied by 44.4 cubic feet of gas.

Each pound of water will require 970 B.t.u. to evaporate it at 212 degrees Fahrenheit. To supply this amount of heat to the boiler

$$\frac{970}{0.75} = 1293\ B.t.u.$$

must be supplied by the gas. If each pound of water requires 1293 B.t.u. it is evident that 100 cubic feet of gas, which contains 100,000 B.t.u., will evaporate 77.3 pounds of water.

W. L. DURAND.

Washington, D. C.

Irrigation Problem

In the "Irrigation Problem" presented by L. J. Robertson in the issue of October 18, he makes no mention of the diameter of the pipe line or of the horsepower required to drive the pump; hence, no definite figures can be given in the explanation. However, simple formulas can be given in which substitution will give the desired results.

Nothing would be gained by the second proposed change as it makes no difference in the operation of the pump as long as its position is below the maximum working atmospheric lift, that is, 26 feet above the water level for small pipes and less with increase in the diameter. The fact that the combined length of the suction and discharge pipes remains the same, the same friction losses are present, provided there is no change in diameter or in the number of bends.

The first proposed change is the one that would result in a decrease in friction and consequently a decrease in the power required to drive the pump.

Friction in pipe lines is made up of several elements. First, a loss due to entrance. This is dependent on the shape of the end of the suction pipe and the velocity, and runs from

$$\frac{0.02\ v^2}{2\ g}$$

for a bell mouth without strainer, to

$$\frac{0.095\ v^2}{2\ g}$$

for a straight mouth without strainer, v being the velocity of the water in feet per second, found by dividing the flow in cubic feet per second by the area of the opening, and g being the value of acceleration due to gravity, 32.2 feet per second. A strainer would materially increase the loss due to increased velocity through the mesh. Second, a loss due to the friction in the straight runs. This depends on the velocity and the diameter of the pipe. It would equal

$$\frac{f\,l}{d} \times \frac{v^2}{2\ g}$$

where

l = Length of pipe in feet;
d = Diameter of pipe in feet;
f = Friction coefficient depending on the nature of the inside surface of the pipe. A common value taken is 0.02.

Third, a loss due to curvature. This would equal

$$\frac{f_c\,l}{d} \times \frac{v^2}{2\ g}$$

where f_c is a coefficient of curvature depending on the radius of the bend and the diameter of the pipe. If R equals the radius of the bend and d equals the diameter of the pipe, as $R \div d$ decreases f_c increases.

$\frac{R}{d}$ =	24	16	10	6	4	2.5
f_c =	0.036	0.037	0.047	0.060	0.062	0.072

In sharp-angle fittings the loss factor is greatly in excess of those just given.

The advantages to be gained by a change in line in accordance with the sketch would be 50 per cent. shorter suction pipe and consequently 50 per cent. less friction loss in the runs, provided the ditch to take the place of this length of pipe be made of such a size that the inflowing water will have a velocity less than one foot per second; disposal of three 45-degree elbows and one 90-degree bend and, consequently, their friction loss less the small loss due to the new 30-degree bend. The entrance conditions are considered to remain the same, hence no added loss or gain at this point.

The total head pumped against would be 22 plus 18½ feet, plus the sum of the different losses given above, plus the loss in the pump, all reduced to feet of head. Since the entrance loss is considered as being constant and the loss in the pump is so, these can be omitted from the discussion. The resultant horsepowers, both under the present system and under the proposed system, are

$$\textit{Present horsepower} = \frac{q \times w \times H}{33{,}000}$$

$$\textit{Proposed horsepower} = \frac{q \times w \times H'}{33{,}000}$$

In the two formulas just given

q = Cubic feet per minute = $\frac{16{,}500}{7.48}$
w = 62.5 pounds;
H = Sum of all heads in present system;
H' = Sum of all heads in proposed system.

The saving in horsepower equals

$$\frac{q \times w \times (H - H')}{33{,}000}$$

This can actually be saved as power or can be used to give a greater output.

D. B. TAYLOR.

Troy, N. Y.

Writers among Engineers

B. P. Gage's article in the October 25 issue, regarding writers among engineers, touches upon a subject that should be given more attention by the practical man. There is rarely an issue of POWER from which I cannot get some useful information. Whenever a subject comes up on which I am not posted, a letter asking for aid will bring me a store of information from practical men that could not possibly be found in a dozen volumes treating the subject technically.

Perhaps a great many men think that what they would write could not interest others. This is a wrong view to take of the matter. It not only interests others but aids the writer to a certain extent, since he will study his subject more closely that he may express himself more clearly. There are very few of the calling who have not directly or indirectly had some aid from others, regardless of their assertions to the contrary. The plan of helping those who help themselves should be followed among engineers. The columns of POWER are a most appropriate place in which to execute this plan.

CARL E. WEBSTER.

Higley, Ariz.

Quarter Turn Drive

Referring to "A Quarter Turn Drive" in the November 1 issue, by Walter J. Bitterlich, a distance of twice the diameter of the larger pulley between centers would be too close with a 6-inch belt for economy. However, a 4-inch belt would give entire satisfaction. If I were to use a 6-inch belt for a drive of this nature, I would at least make the distance between the centers of the pulleys two and one-half times the diameter of the largest pulley.

Regarding the maximum belt width it is well known that a thick, narrow belt is better than a thin, broad one, being more durable and working more satisfactorily.

The belt speed for maximum economy should be from 4000 to 4500 feet per minute.

T. J. HAMMERSLEY.

Milwaukee, Wis.

The Problem of Smoke Abatement

Discussion of a paper delivered by D. T. Randall at the Boston meeting of the American Society of Mechanical Engineers on November 10, an abstract of which appeared in our November 15 issue.

After the presentation of Mr. Randall's paper on "Smoke Abatement" at the joint meeting of Boston engineers, on November 10, an interesting discussion took place. Mr. Bailey was first called upon and his remarks in abstract follow:

E. G. BAILEY—The essential factors which enter into the prevention of smoke have been known for many years, yet the economical smokeless combustion of coal in everyday practice is still far from being attained. Although the Ringelmann smoke chart has been used for many years, it has only recently received the attention which it deserves. Its use has been greatly stimulated through its adoption and reproduction in convenient form by the United States Geological Survey.

The personal factor of the observer enters into the readings with the Ringelmann chart to a certain extent, but this causes a much smaller error than is generally believed. The greatest difference between the readings of different observers is due to the smoke being sometimes of a brownish cast as compared with the jet black ink used in making the charts. If the charts were made the same shade as the smoke, the chance for personal error would be lessened. However, when using the black chart one should remember that smoke corresponding to No. 1 must allow 80 per cent. of light to pass through it; No. 4, 20 per cent.; and No. 5 is opaque, regardless of the exact matching of color shades. When comparing smoke with the chart, the smoke should be as dense as, or denser than the number assigned it, yet not so dense as the next higher number. For instance, if smoke is denser than No. 3, and not so dense as No. 4, it should be recorded as No. 3. Likewise, if some smoke is being emitted from a stack, yet it is not so dense as No. 1, it should be recorded as — 1 or + 0.

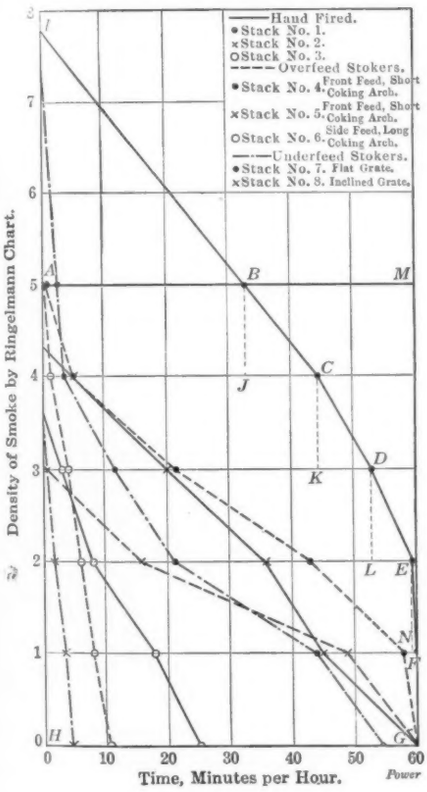

Hand Fired.
Stack No. 1.
Stack No. 2.
Stack No. 3.
Overfeed Stokers.
Stack No. 4. Front Feed, Short Coking Arch.
Stack No. 5. Front Feed, Short Coking Arch. Side Feed, Long Coking Arch.
Stack No. 6. Coking Arch.
Underfeed Stokers.
Stack No. 7. Flat Grate.
Stack No. 8. Inclined Grate.
Density of Smoke by Ringelmann Chart.
Time, Minutes per Hour.
Power

FIG. 1.

Ringelmann chart readings are usually taken at half-minute or minute intervals during the period of an hour or more in the case of stationary stacks, and the readings are plotted in the form of a log, or the density of the smoke is obtained by averaging all readings and expressing the result in percentage of smoke on a basis of No. 5 being 100 per cent. The plotted continuous log gives a good general idea of the manner and regularity of the smoke emission, but is very unsatisfactory for comparing one stack with another, or the same stack from time to time. The average of the readings alone gives but little idea of the nature of the smoke, whether continuous and of low density, or in intermittent puffs of denser smoke. Another method which I have used with good results for several years is to plot a series of readings as one characteristic curve. This is done by adding the total number of readings of each different density, reducing them to a basis of minutes per hour, adding to the number of minutes corresponding to each density all of the minutes corresponding to the greater densities, and plotting Ringelmann chart numbers as ordinates and minutes as abscissas. For example, with stack No. 7, Fig. 1, the readings of a two-hour period reduced to minutes per hour are given in Table 1.

The values in the last column are plotted as abscissas on the lines corresponding to the different density numbers, and the curve drawn through these points indicates the general character of the smoke being emitted.

The curves in Fig. 1 represent a variety of plants located in different places with different equipments, but all burning semi-bituminous coal containing between 17 and 21 per cent. of volatile matter. Most of this data was obtained in connection with the drafting of the smoke bill recently enacted for Boston through the efforts of the fuel-supply committee of the Boston Chamber of Commerce.

No. 5 on the Ringelmann chart includes all smoke that is opaque, and, for a stack of a given diameter, a certain amount of carbon particles per cubic foot is necessary to make this smoke opaque, but twice as much carbon or smoke could be carried per cubic foot of gas without changing the reading. This point is graphically brought out in these characteristic smoke curves. For instance, stack No. 1 would not be apt to produce smoke of No. 5 density, no more nor no less, for thirty-three minutes out of an hour; but during a part of this time it was undoubtedly much denser, as indicated by carrying the curve up until it intersects the zero line at a density of about 8. The exact nature of the extended part of this curve is problematical, but it illustrates the fact that the average density of smoke may be much greater than is indicated by merely averaging the readings in the usual method. The average density of the smoke, as compared with No. 5 as a basis, is the area *H I B C D E F G H* divided by *A M G H A*, or 103 per cent.

TABLE 1.

Density, Ringelmann Chart Number.	Time during which smoke of each density was emitted. Minutes.	Time during which smoke of each density and darker was emitted. Minutes.
5	2.5	2.5
4	1	3.5
3	8	11.5
2	9.5	21.0
1	23	44.0
−1 + 0	10.5	54.5
0	5.5	60.0
Total.....	60.0	....

If the area representing smoke denser than No. 5 were neglected, the result would be area *A B C D E F G H A* divided by *A M G H A*, or 87.8 per cent. If the usual method of averaging smoke readings, such as was apparently used in the United States Geological Survey bulletin No. 373, be applied to this stack, the density referred to No. 5 is only 83.3 per cent. This error is due to the omission of the triangular areas *B J C B*, *C K D C*, *D L E D* and *E N F E*, which properly belong in the area representing the total smoke emitted. All of the No. 4 readings, representing the eleven and one-half minutes from *J* to *C*, were as dense or denser than No. 4, and less than No. 5, so they would naturally be distributed along the diagonal line *B C*, and the same holds true with respect to the other parts of the curve. Applying the averaging method to the curve representing stack No. 5, Fig. 1, gives 21.8 per cent., although the true average from the area

is 31.8 per cent. The flatter the curve, the greater is the error caused by the use of the averaging method as usually applied, and the result is always too low.

Stacks Nos. 4, 5 and 6 were connected to stoker furnaces of the over-feed type. Nos. 4 and 5 had short coking arches with front feed, and the plants were similar with respect to the number of boilers and the size of stack, but the relative rates of combustion, volatile matter in the coal, etc., are not known. Stack No. 6 had only one furnace of the side-feed type with a long coking arch connected to it. Stacks Nos. 7 and 8 were connected to under-feed stokers, the former having five stokers with flat grates, and the latter three stokers of the inclined-grate type.

No specific conclusions as to the value of the different methods of firing should be drawn from the curves of Fig. 1, as they are merely given to show the great variation that exists in actual practice between plants with similar equipment.

TABLE 2.

Stack Number.	Diameter of Stack, Feet.	Number of Boilers Connected.	Type of Boilers.	Rated Boiler Horsepower.	Grate Area, Square Feet.	Method of Firing.
1	8	6	Horizontal Return Tubular.	900	204	Hand spreading
2	7½	8	Horizontal Return Tubular.	1200	272	Hand spreading
3	4	1	Porcupine.	300	56	Hand spreading
4	7	1	Horizontal Return Tubular.	1050	238	Hand spreading

TABLE 3.

	C	D
Analysis of coal as fired:		
Moisture, per cent	4.81	5.40
Volatile, per cent	16.22	18.81
Fixed carbon, per cent	70.40	69.00
Ash, per cent	8.57	6.79
Sulphur, per cent	1.20	0.70
B.t.u.	13,639	13,838
Average draft in firebox, inches of water:		
Boilers of stack No. 1	0.45	0.32
Boilers of stack No. 2	0.25	0.31
Boilers of stack No. 3	0.30	0.32
Boilers of stack No. 4	0.21	0.27
Coal fired per square foot of grate per hour, pounds:		
Boilers of stack No. 1	15.1	13.8
Boilers of stack No. 2	11.9	11.8
Boilers of stack No. 3	19.6	16.5
Boilers of stack No. 4	9.2	11.7
Flue-gas analysis, percentage of CO_2 by volume:		
Stack No. 1	8.1	7.7
Stack No. 2	6.9	7.4
Stack No. 4	7.4	9.5
Flue-gas analysis, percentage of CO by volume:		
Stack No. 1	0.0	0.0
Stack No. 2	0.0	0.05
Stack No. 4	0.0	0.20
Excess air from flue-gas analysis, per cent:		
Stack No. 1	137.00	146.00
Stack No. 2	180.00	149.00
Stack No. 4	155.00	87.00
Percentage of smoke with No. 5 on Ringelmann chart as 100 per cent., from Fig. No. 3:		
Stack No. 1	13.8	22.8
Stack No. 2	18.2	35.8
Stack No. 3	34.7	56.0
Stack No. 4	4.3	17.2

The curves shown in Fig. 2 were all taken from the same stack connected to one internally fired boiler of the locomotive type. The same kind of coal was fired in each test by the same fireman, using the spreading method. The only changes made on the different tests were the amount and the temperature of secondary air supply. The coal used on these tests contained about 30 per cent. volatile matter, and over 50 pounds of it were burned per square foot of grate surface per hour.

The agreement between different observers is shown in tests Nos. 1 and 3. Observers *A* and *C* had had very little experience in the use of the Ringelmann chart, and did not know in either case that observer *B* was taking readings.

The curves of Fig. 3 represent the smoke emitted from four different stacks at one plant on two eleven-hour tests made on consecutive days. The tests were primarily made to determine the comparative values of the different coals from the evaporation on the entire plant, and the smoke readings were taken as a side issue, as the location of the plant was such that the prevention of smoke was given no consideration. There were nine firemen, and their methods of firing were not changed in any way from their usual practice.

Table 2 gives the general dimensions and equipment connected with each stack, and Table 3 contains the general data which have direct bearing upon the amount of smoke produced. All conditions were maintained as nearly uniform on the two tests as the daily operation of the plant would permit, the only intended change being in the coal burned. The principal difference between the coals so far as smoke was concerned was in the percentage of volatile matter, which was 16.22 and 18.81 in coals *C* and *D*, respectively, as fired, or 17.04 per cent. and 19.88 per cent. on the dry basis. Owing to changes in the draft and the amount of clinker formed, the rate of combustion was not quite the same for the two tests. There was also some change in the flue-gas analysis and air excess. The gas samples were taken from the main flues, and included considerable leakage of air through the boiler settings.

Slight differences in the percentage of volatile matter, rate of combustion, excess air, method of firing, etc., are found to have a marked effect upon the amount of smoke produced when the characteristic curves are drawn. By the use of such data and curves, it is possible to determine the relative importance of the different factors affecting this problem, and to take intelligent steps to reduce the density of the smoke to the desired limits.

FREDERICK H. KEYES—The power plants of New England are so situated with reference to the source of supply that, as a rule, it is impracticable to use anything except bituminous coal; and unfortunately the price is usually the controlling factor rather than quality, and I believe that the quality is largely responsible for the amount of smoke. Some of the largest users of bituminous coal have already found it advantageous to buy their coal under specifications especially drawn up to meet their requirements; and this would seem to be the first step in the right direction, so far as smoke is concerned.

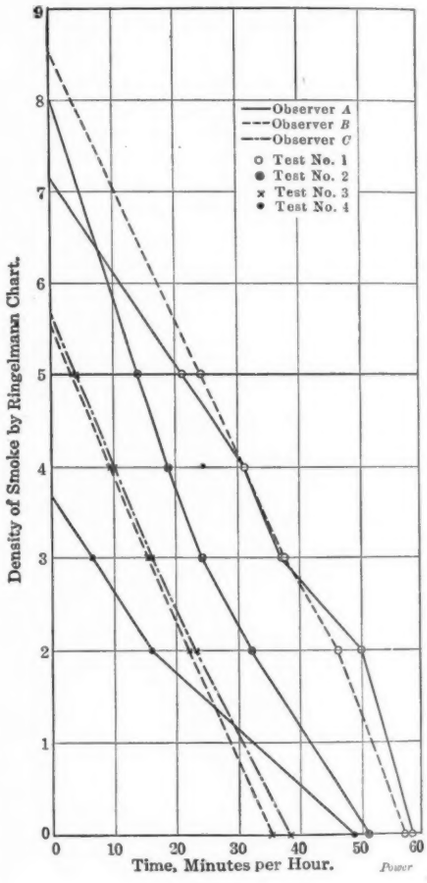


FIG. 2.

Having obtained coal with the minimum of smoke-producing qualities, the next step is to fire and burn it properly. Here is where it is absolutely necessary to have the coöperation of the fireman.

Next to the fireman, the quantity of air, together with the manner and conditions under which it is mixed with the products of combustion, is of the greatest importance; but this point must be determined for each individual case according to the kind of coal used and the

general conditions governing the operation of the plant.

I believe that special devices, such as steam jets and firebrick arches, with few exceptions, usually cost more to operate and maintain than can be saved through the use of low-grade fuel for which such special devices are invariably required to obtain even approximately satisfactory results.

HENRY BARTLETT—I have read with much interest Mr. Randall's paper and note that he deals with smoke abatement from a stationary standpoint. I shall confine my remarks to the problem of smoke prevention with locomotives. Bituminous coal is the principal fuel supply for locomotives, and the subject of smoke abatement has been a live one, especially

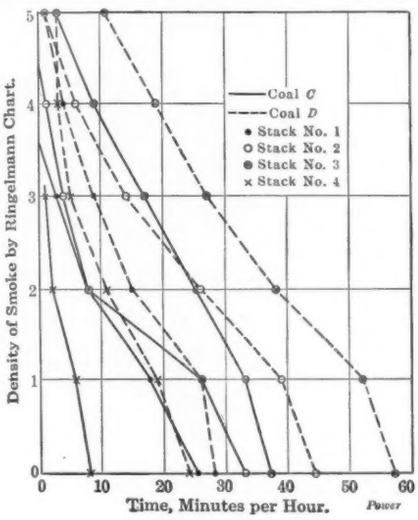


FIG. 3.

within our cities, where the situation has become very serious.

The locomotive boiler is called upon for ranges of work greater than that of any other service. One moment the locomotive is standing quietly in the roundhouse with banked fire and only 40 or 50 pounds of steam on the boiler, and within the hour it may be running on the road at 200 pounds of steam and exerting its maximum power. It may be moving along pulling a heavy train and emitting little smoke, when the next moment it is suddenly stopped or required to take a siding. Under such changes it is impossible to eliminate the smoke as the large body of coal necessary to perform the work keeps on emitting the volatile gases until these have disappeared. The terminals of runs are necessarily in large cities and communities where locomotives are housed and have their fires cleaned and built up for their runs.

To avoid this the question might be asked—why not burn smokeless fuel? I only wish this solution of the problem were possible, but the only smokeless fuels available are anthracite coal, coke and oil. Anthracite coal is found principally in Pennsylvania, where statistics show that 70,000,000 tons are mined annually, or less than the amount of coal yearly consumed by the railroads. Considering the demands for this fuel in domestic purposes, where no other fuel seems available as a substitute, should the railroads resort to its use in general, the supply would be insufficient and the price would so advance as to render the cost prohibitive.

As regards coke the Boston & Maine railroad is more fortunately situated in having on its line a large producer, and for the last few years has availed itself accordingly to the fullest practical extent.

Oil properly fired is a smokeless fuel, found extensively in a region where it is largely used by the neighboring railroads, but it also is limited in supply and the oilfields are so located that the cost of its transportation would make it prohibitive for general use.

Smoke-consuming devices have been largely employed by different railroads in trying to solve this momentous smoke problem and large amounts of money have been spent in seeking the ideal in this direction, but so far without success. The only device of this kind which is retained in common use is the brick arch.

GEORGE H. BARRUS—When a charge of bituminous coal is thrown into a hand-fired furnace, the first thing which occurs is the heating of the coal to the temperature of ignition. During this process the volatile gases in the coal are driven off. These escape with great rapidity in the early stages, after which the quantity of gas gradually diminishes, and finally nothing but incandescent coke remains.

The gases first evolved from a charge of coal are in their original cold state. Contact with the radiant heat of the furnace increases the temperature of the exposed portions of the gases sufficiently to ignite them. Those portions of the gas which are not favorably exposed receive an insufficient quantity of heat to become thoroughly ignited, and these pass out of the furnace in an unconsumed state. It is the underheated gas thus formed which causes smoke, and the whole problem of smoke prevention consists in heating sufficiently these gases before leaving the combustion chamber. When they receive the proper amount of heat before entering the boiler, they are burned without smoke and the degree of smokelessness obtained depends upon the degree with which the heating of the gases has been effected.

This feature of the subject may be illustrated by referring to an experiment I made on a battery of two 300-horsepower vertical-pass horizontal water-tube boilers. The result aimed at was to ascertain the effect of firing the two boilers alternately and employing the incandescent coke of one furnace as a medium for overheating the gases evolved from fresh charges of coal in the other furnace.

To prepare for the experiment an opening was cut through the intermediate wall between the two furnaces. This opening, which extended horizontally the whole length of the grate and vertically from the grate to the lower row of tubes, put the two furnaces into free communication, and the closing of either of the flue dampers was sufficient to cause the products of combustion from the furnace of one boiler to pass through the intermediate opening and over the fuel bed of the other furnace, and thereby secure the object sought.

The method of operation was as follows: Designating the boilers as Nos. 1 and 2, the two furnaces were fired alternately at equal intervals, and whenever a boiler was fired its damper was closed. The products of combustion from No. 1 furnace, just fired, passed through the intermediate opening, over the fuel bed of No. 2 furnace, mixed with the products of combustion from that furnace and passed on through No. 2 boiler and No. 2 damper. When the volatile gases had distilled off from the fresh coal, the positions of the two dampers were reversed, No. 1 being opened and No. 2 closed. Then No. 2 furnace was fired and the volatile gases from this furnace passed through the intermediate opening into No. 1 furnace, thence over the incandescent fuel bed in that furnace and on into No. 1 boiler along with the products from No. 1 furnace.

To compare the amount of smoke produced in this system with the ordinary system of operation, the boilers were first run in their usual manner. New River coal was used and was charged through the two outside doors first, there being three doors in all for each boiler, and then after waiting three to five minutes, the middle door was charged, one boiler being fired at a time. Smoke observations were made every minute for a continuous period of two hours, the estimated percentage of dense black smoke being judged as it appeared to the eye on escaping from the top of the chimney.

The average density of smoke observed for the entire period of two hours, including the time when there was no smoke, was 13.6 per cent., while the maximum was 75 per cent.; and there was an entire absence of smoke, or only a trace, for fifty-four minutes.

When the alternate system was brought into use, the three doors of each boiler were fired in rotation, there being no waiting between doors. In general, 12 shovels of coal were fired at each round.

The average density of smoke observed during the entire period of two hours, including the time of no smoke, was 2.3 per cent. The maximum amount was 10 per cent., and there was an entire absence of smoke, or only a trace, for one hour and twenty-two minutes out of the two hours.

Inquiries of General Interest

Number of Expansions

How may I find the number of expansions taking place in the cylinder of a steam engine?

N. O. E.

The number of expansions taking place in a steam-engine cylinder is always the reciprocal of the cutoff; that is, one divided by the cutoff. If the cutoff is one-fifth, the number of expansions will be five. The number of expansions may also be found by dividing the absolute initial pressure by the absolute terminal pressure. If the initial pressure is 160 pounds gage, 175 pounds absolute, and the terminal pressure is 20 pounds gage or 35 absolute, dividing the former by the latter will give five as the number of expansions.

Weight of Safety Valve Lever

In safety-valve calculations, why is the distance of the center of gravity of the lever from the fulcrum taken?

W. S. L.

The weight of the lever acts on the valve, tending to keep it closed against the pressure of the steam, and the effect is the same as it would be if its total weight were concentrated at the center of gravity; therefore, the distance of the center of gravity of the lever from the fulcrum is multiplied by its weight.

Pulley Dimensions for a Given Horsepower

What determines the size of a pulley needed to transmit a given horsepower?

P. D. H.

The power to be transmitted and the speed at which it revolves. For a single belt choose a diameter and face that for the number of revolutions will give from 60 to 70 square feet of belt surface passing over the pulley per minute for each horsepower. For a double belt the surface may be reduced one-fourth.

Setting Engine Valves

What is the correct rule for setting the valves of Corliss engines?

S. E. V.

The elements of valve setting, whether for Corliss or other engines, may be stated in a very few words. But each step in the process may be enlarged upon almost without limit. No rule, however elaborate, will furnish anything beyond directions to perform certain operations. The briefest will do as much. To set the valve or valves of any engine, make the eccentric and valve rods of the right length, then with the engine on the center turn the eccentric in the direction the shaft is to revolve until the proper amount of lead is secured; fasten the eccentric and the job is done.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Steam Chest Pressure

What should the steam pressure be in the steam chest of an engine with a throttling governor, the engine running with a normal load and the boiler pressure being 100 pounds?

H. B.

Assuming a mean effective pressure of 40 pounds in the cylinder and a cutoff at five-eighths of the stroke and that 80 per cent. of the theoretical pressure is realized, then

$$40 \div 0.80 = 50 \textit{ pounds},$$

the theoretical mean effective pressure. At five-eighths cutoff the mean effective pressure will be nine-tenths of the initial pressure. If the mean effective pressure is 50 pounds the initial will be 55.55 and as the probable fall between the steam chest and the cylinder will not exceed two pounds the steam-chest pressure may be taken to be

$$55.55 + 2 = 57.55 \textit{ pounds}.$$

Width of Exhaust Port

In laying out a valve seat, how can I determine the correct width of the exhaust port?

W. E. P.

The width of the exhaust port for a plain slide-valve seat should be the width of the steam port, plus one-half the travel of the valve, minus the width of the bridge between the steam and the exhaust ports.

Pressure in Feed Pipe

In order to feed water into a boiler carrying 80 pounds pressure, must the pressure in the feed pipe be above that in the boiler?

P. F. P.

Yes. In order to produce a flow in any system there must be a difference in pressure. If the pressure is the same in the pipe as in the boiler there will be no flow. If the pressure in the boiler is above that in the pipe, the flow will be from the boiler into the pipe, and if greater in the pipe than in the boiler, the flow will be from the pipe to the boiler.

Temperature of Steam at Reduced Pressure

I have a vulcanizer using steam reduced to 40 pounds from a pressure of 100 pounds. What is its temperature?

T. R. P.

The total heat in steam at 100 pounds pressure is 1188.8 B.t.u.; at 40 pounds, 1175.4

$$1188.8 - 1175.4 = 13.4 \textit{ B.t.u.}$$

The specific heat of steam at 40 pounds pressure is approximately 0.52; therefore, the additional 13.4 B.t.u. in the steam at the reduced pressure would superheat it 25.7 degrees.

The temperature of steam at 40 pounds pressure is 287.1 degrees and the 25.7 degrees added will make it 312.8. This is assuming that the steam at 100 pounds pressure is dry. It may contain so much moisture that when reduced to 40 pounds its temperature will not be above that due to its pressure.

Efficiency of Boilers

What is the difference in efficiency of a boiler when expressed in pounds of coal and pounds of combustible?

E. O. B.

The heat absorbed by the boiler is divided in both cases by the calorific value of the fuel.

In one case the efficiency is expressed by the formula,

$$\textit{Efficiency} = \frac{\textit{Heat absorbed per pound of combustible}}{\textit{Calorific value of one pound of combustible}}$$

and in the other by

$$\textit{Efficiency} = \frac{\textit{Heat absorbed per pound of coal}}{\textit{Calorific value of one pound of coal}}$$

Advantages of Compound Engines

What are some of the advantages in using a compound engine rather than a simple one?

A. C. E.

A higher initial pressure, a greater number of expansions and less cylinder condensation, owing to a shorter temperature range in each cylinder than would obtain if the whole number of expansions took place in one cylinder.

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November 29	34,000

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Contents

PAGE

Compound Engine Proportions..... 2094
Milwaukee Garbage Destructor..... 2097
Factors Affecting Air Pump Capacity... 2100
The Confession of an Engineer..... 2103
A Boiler Efficiency of 82.36 per Cent... 2105
An Engine Built up of Sections..... 2106
Charging a Refrigerating System..... 2107
Trouble with Water Column Piping.... 2108
Care of Rotary Converter Commutators 2109
Direct Current Motors and Their Connections 2109
Central Station Growth..... 2110
The Burke Induction Motor..... 2111
Flying Machine Motors..... 2112
A Puzzling Case of Premature Ignition 2114
Corrosion of Exhaust Pipe Drips..... 2114
Cracked Flywheel Hubs..... 2114
Practical Letters:
Power Plant Design and the Operating Engineer .... Gas in the Pump....High Water Alarm....A Convenient Flue Cutter .... Color Scheme for Piping....A Homemade Oiling System.....2115-2117
Discussion Letters:
The Theory of the Cooling Tower ....Right Angled Triangular Weir ....Indicator Reducing Motion.... Leakage through a Piston Valve .... One Valve Performing the Functions of Two....Firing with Natural Gas....Irrigation Problem ....Writers among Engineers.... Quarter Turn Drive.....2118-2122
The Problem of Smoke Abatement..... 2123
Editorials 2127-2128

Power and The Engineer

Editorial

Accumulators for Furnace and Boiler Capacity

It is a well recognized fact that a steam engine of given displacement will operate most efficiently upon a particular load under given conditions of initial and back pressure. The engine which drives a vessel, a pump or a cotton mill has a decided advantage, if it is at all well proportioned to its work, over the engine which supplies the varying demands of an electric railroad, and which can be running at its maximum efficiency only a small part of the time.

This is equally true, if not more true, of the boiler. A boiler can be designed to furnish a given amount of steam per hour in which both the furnace and the boiler efficiency will be high; but, if this boiler has to work at one moment fifty per cent. above the capacity to which it is best adapted and at another at fifty per cent. less, the result must be less favorable.

It is sought to ameliorate the engine condition by straightening out the load curve as much as possible. A storage battery of sufficient capacity would do this completely, but would cost more than the saving would come to. The capacity is also adjusted to the load, when the variations are slow enough and of sufficient amplitude, by cutting in and out engine units.

The boiler condition may be ameliorated in the same way either by cutting in or out more or less units or by providing some sort of an elastic takeup or accumulator between the boiler and the engine.

The steam space, including that of the piping, furnishes such a takeup to a limited extent. When the demand exceeds the rate at which the boilers are working the steam space becomes partially depleted and the pressure falls. If, on the other hand, the rate of demand is less than that at which the boiler is working, the pressure will rise.

It has been one of the first precepts of a boiler room to keep the pressure constant, but it is a question if the boiler pressure cannot be allowed to vary through a considerable range with less damage to the over-all efficiency than would result from the constant manipulation of the damper and the slice bar necessary to hold it constant. At the pressures ordinarily carried, a considerable pressure drop will produce a comparatively insignificant change in the initial temperature, and it is the temperature range which affects the efficiency.

Some years ago there was introduced in England a system of thermal storage. Great tanks in the boiler room were filled with water which was allowed to circulate in the boilers and become heated when the load was light, and which in its heated condition was drawn upon when the load was heavy. It is doubtful if the possibilities of such a system, combined with economizers, have been exhausted.

The boiler furnace is at the same time a gas producer and a gas burner. If it gets to producing gas faster than it is needed to be burned, there is trouble. Sometimes there is trouble to burn, in the constricted combustion chamber, all the gas which it can produce. The dutch oven or extension furnace is a move in the direction of a separate producer and a gas-fired furnace. Something like a gasometer between the producer and the furnace would allow the producer to work always at its maximum efficiency and the combustion to be controlled in accordance with the load, but such an outfit would probably bear the same relation to the boiler that the storage battery does to the engine, and if gas were available it would be better to burn it in the engine itself rather than in the boiler.

The limit of boiler capacity seems to lie not in how much heat the surfaces can absorb, but in how much coal can be burned in the furnace. It is largely a question of how much firebrick one can afford to burn up as against buying and running more boilers.

Proximate Analysis of Coals

The present tendency in power-plant operation is to pay special attention to the quality of fuel purchased, not only as to its total heat contents, but also for the purpose of determining the proportions of its various constituents so that it may be adaptable to the service required. Most of the large plants have facilities for making accurate analyses of the coal received, such tests forming part of the regular operating routine; but this necessitates a somewhat elaborate apparatus and the employment of an expert to make the tests, the expense of which, in the large plants, is more than compensated for by the benefits derived.

In the small plants, however, it is not always expedient to go into such refine-

ments; at the same time, it is desirable that a check should be made upon the quality of coal being supplied. For such purposes the proximate method of analysis may be resorted to. This gives the approximate proportions of fixed carbon, volatile matter, moisture and ash. It is made by subjecting a sample of coal to a temperature of 250 to 300 degrees Fahrenheit to drive off the moisture; then to a red heat to drive off the volatile matter; then to a white heat which causes the carbon to pass off as carbon dioxide, leaving the ash. By weighing the coal in the beginning and the residue after each operation the various percentages can be computed. By exercising a little care most anyone can make such an analysis; and, although it does not afford an accurate basis for computing the heat value, it does afford information as to the general characteristics of the coal and also a means for comparing the relative heat values of various coals.

The Human Element

Not long ago an analysis of the cause of all of the accidents reported from power plants for one year disclosed the fact that over seventy per cent. were due to the carelessness of attendants. Human machines are, according to the result of this investigation, less than one-half as reliable as those made of wood and metal.

To what extent the mental and physical condition of the human element influenced the result will probably not be known; but it would be interesting to learn for a certainty whether the man with tasks well within the limits of his mental and physical strength is more prone to neglect obvious duties than the one with hours and toil beyond the ability of the average man.

Is the overworked fireman more liable to allow an erratic water level in the boiler than the one with time for smokes and naps between firings?

Is the oiler watching a single engine more attentive to the hight of the oil in a few cups or more solicitous as to the temperature of the bearings than he who can scarcely replenish the last cup and touch the swiftly passing crank pin before the first one is empty?

There are engineers, firemen and oilers the routine of whose uneventful daily work is never varied by an emergency, and there are others whose utmost energy is taxed in replacing broken follower bolts just in time for the heavy run while a blown-out joint in the main steam line shrieks for attention.

In the great majority of power-plant accidents is it the man or the material that is primarily at fault?

To the personal equation the analyst has ascribed seventy per cent. of the failures. But in the remaining thirty per cent. due to faulty design or construction, how many of the accidents could have been avoided if the mind of the man had been on his work? An engineer who is alert and resourceful is a treasure, but one with that forethought and attention to operating conditions which renders exhibitions of spectacular resourcefulness unnecessary is priceless.

Gas Engine Valves

It may not occur to many operators of gas engines that the heads of the inlet and exhaust valves have to withstand exactly the same pressure per square inch that is applied to the other interior surfaces of the combustion chamber when the charge is ignited. Of course, it is obvious when one is reminded of it, but a good many inexperienced designers have made the mistake of regarding the valves merely as hole-closers and have made the disks thin and the heads flat to reduce the weight and the consequent tendency to hammer the seats. The unsupported area of a three-inch valve disk is about seven square inches, and when the explosion pressure goes up much beyond three hundred pounds the total pressure on such a valve is more than a long ton. That is why most valve heads are heavily crowned.

The proper way to prevent the valves from hammering their seats is to make the retreating curve of the lifting cams such that the cams won't "run away" from the rollers and allow the springs to bang the valves down on their seats. This sounds like the essence of simplicity, but it isn't. The proper layout of gas-engine valve cams comes pretty near being a fine art.

Enlightening the Engineer

Not long ago the subscription representative of a leading engineering paper called at the office of a large manufacturing concern, and upon stating his business to the superintendent he was told that the firm had no use for such a thing as a technical paper. Thereupon, he requested permission to talk the matter over with the engineer, to which the superintendent replied: "Positively no, I do not allow my engineer to fool with such stuff; and, furthermore, if I hear of him doing so, I will fire him."

Thus launching into the subject he went on to say that he always knew what was needed and how the plant should be run without requiring any suggestions from his engineer; besides, his own time was taken up with the commercial side of the business and he had no time to devote to the study of engineering subjects. He claimed that such matter as is contained in an engineering paper only tends to ruin an engineer, giving him big ideas and making him less useful to his employer.

Here is the case of a man condemning himself by his own words. He is under the delusion that the interests of the firm will be served best by keeping his engineer in ignorance of the latest engineering developments, admitting at the same time that he, himself, has no time for such matters. Fortunately for the industrial interests, this type of superintendent is fast disappearing, and is giving way to the wide-awake management which realizes that the engineer plays an important part in the cost of production.

To meet increasing competition in the manufacturing industries, improved machinery has been brought into use and the methods of production have been standardized to such an extent that the zenith of economy has almost been reached in this line. To attain further economy, attention has been directed toward the power plant; the consulting engineer has been called in and new equipment has been introduced which is designed to meet the special requirements of the service. But having gone this far is not enough; the engineer is a factor to be reckoned with and upon his efforts and ability largely depends the efficiency of the plant. For this reason the management should see to it that he is given every opportunity to improve his knowledge and to keep abreast of the times.

Boiler Tubes

News comes of another boiler-tube failure. They are becoming woefully frequent. What is the matter?

Charles S. Blake, of the Hartford Steam Boiler Inspection and Insurance Company, called attention to this condition at the last meeting of the American Boiler Manufacturers' Association. The tubes which are furnished now, he said, for pressures of one hundred and fifty and one hundred and seventy-five pounds are identical in thickness with those that they used to furnish twenty-five years ago, when pressures were from eighty to one hundred pounds. The average quality has not improved. The standard commercial tube is no better than, if as good as, the tubes of those days. We know of two cases where pieces the size of one's hand have come bodily out of tubes as clean as though they had been taken out with a punch. There is something wrong with the making of a tube which fails in that way.

There is a wave motor running at Atlantic City which actually generates electric current and compresses air. Whether it will compete with present methods depends upon the required investment per horsepower. The projectors claim that this cost, including the cribwork, compares favorably with that of a steam plant.

The Climax boiler which exploded at the works of the American Manufacturing Company in Brooklyn, an account of which appeared in our November 1 issue, is claimed to be the first of this type that has ever exploded.

How many books do you know well?

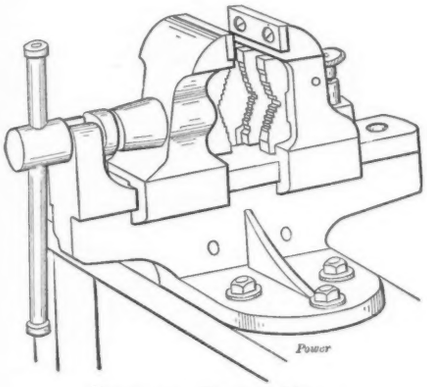
New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Combination Pipe and Bench Vise

The accompanying illustration represents an improvement in a quick-adjusting combination pipe and bench vise. It is provided with an adjusting pin attached to the back jaw of the vise in such a way that it engages in the base of the vise.

This base has three holes in which the pin fits. When the pin is in the first hole, the vise holds from ½- to 1-inch pipe; when in the second, it holds up to 2 inches, and when in the third hole, from 2½- to 3-inch pipe.



PIPE AND BENCH VISE

The vise proper has steel-faced jaws. Sockets for 1¼-inch pipe legs are also provided.

This vise is manufactured by the Armstrong Manufacturing Company, of Bridgeport, Conn.

The Improved Taylor Stoker

In the article on "Under-feed Automatic Stokers" which appeared in the November 15 number, some of our readers may not have realized that the gravity under-feed stoker described was the Taylor, made by the American Ship Windlass Company, of Providence, R. I. Such is the case and since this article has been published, some improvements in the stoker have been perfected.

These changes relate principally to the operation of the lower line of plungers, which push the burned-out coal onto the dumping plates. The travel, or length of stroke, of the upper row of rams which assist gravity by pushing the coal outward into the fire is constant, but the travel of the lower row can be increased to push the refuse out more rapidly just after the dumping plates have been dropped, or whenever it is desired to more quickly get rid of the refuse which has collected in the lower part of the furnace.

As previously built, each lower ram was connected directly to the bell crank which operated the upper plunger. Adjustment of the stroke of the lower plungers was made for each separate retort. With the improved mechanism, the travel of all the lower rams may be lengthened or shortened simultaneously. This is accomplished by means of a sort of pantograph lever connection having a cam and balance weight on the upper member, as shown in the sectional view of the furnace. Pivoted to the plunger connecting rods these cams are connected by means of chains, lever arms and shaft to the operating handle at the center of the stoker. To alter the travel of the lower rams the adjusting lever is dropped, releasing the cams and permitting them to become horizontal, which action lengthens the stroke of the plunger. In case the vertical connection of the bell-crank lever is in such position that the cam cannot drop, it merely remains at rest until the connection moves backward and the counterbalance weight falls and carries the cam into position.

Another change which simplifies the operation of the stoker is the relocation of the speed-shaft sprocket. It is now placed at the end of the stoker with the clutch and faceplates which hold the shearing pin on the outside. This greatly facilitates the replacing of the shearing pins when required and also the throwing of the stoker on or off the line. It also improves the connection to the fan shaft.

A third improvement is the alteration of the dumping plates. In early forms,

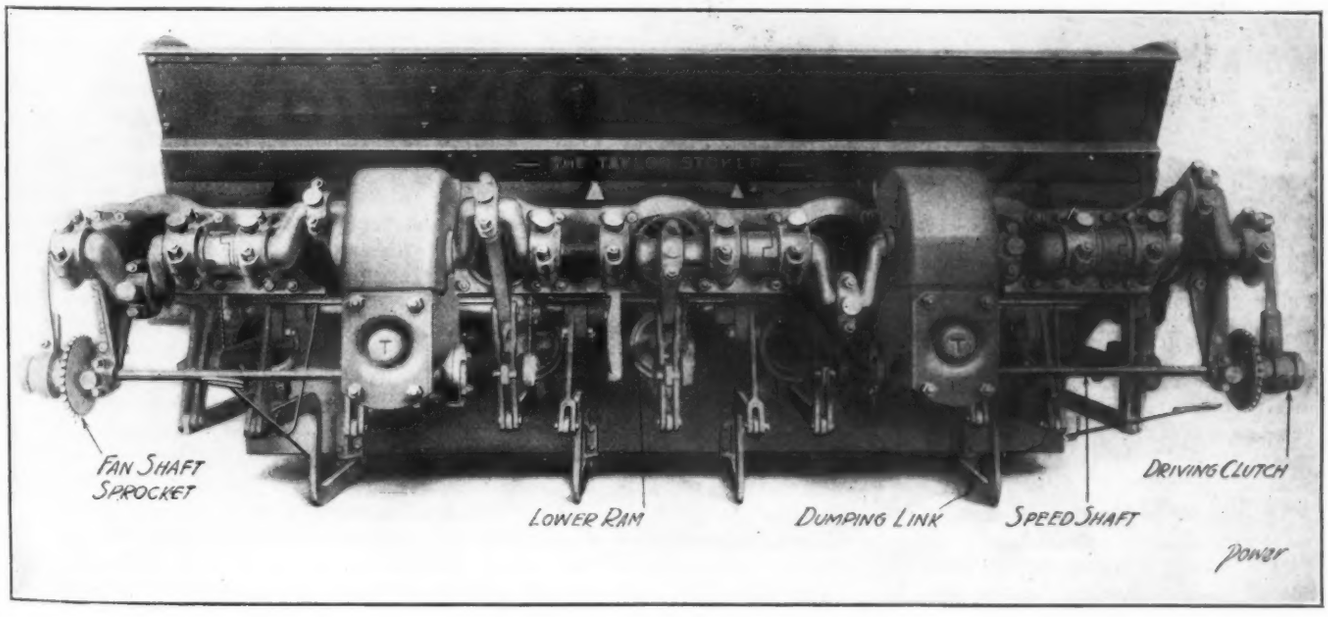


FIG. 1. FRONT VIEW OF TAYLOR STOKER

the dumping plate was cast in one piece but the present design calls for a dumping plate in sections which are independently renewable, facilitating and lessening the cost of repairs. Improvements of minor importance are the use of lock nuts in all places where possible jar

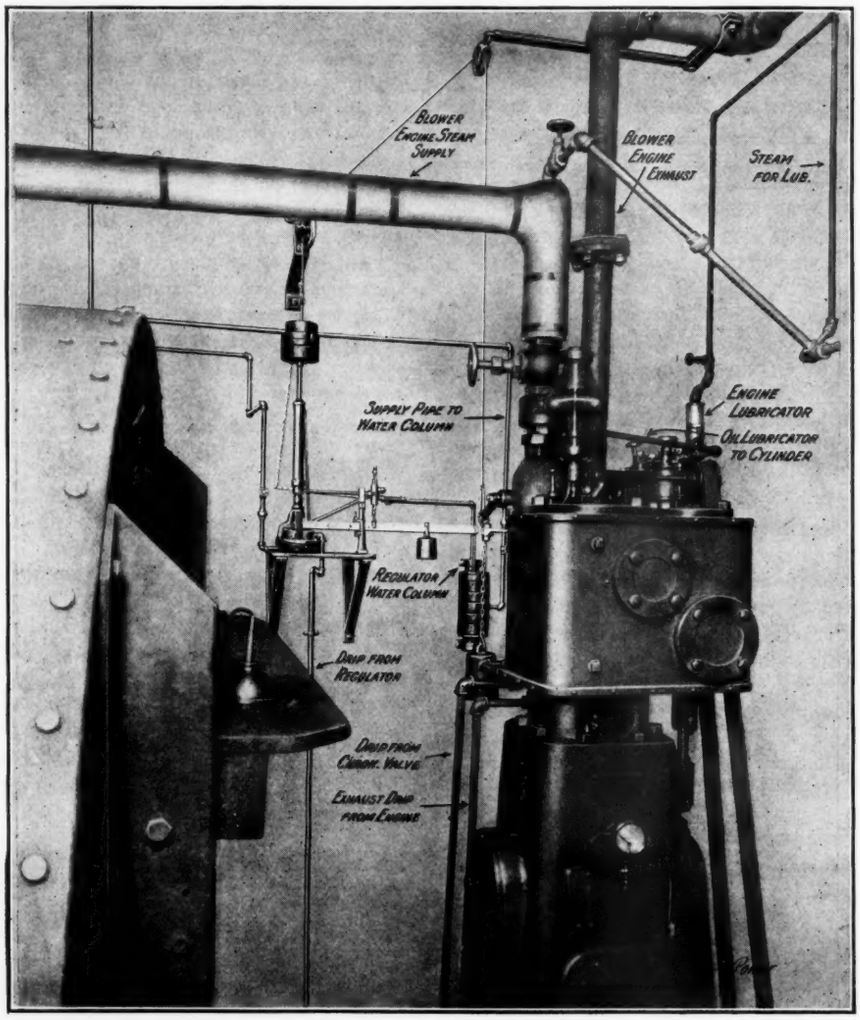


FIG. 2. STOKER ENGINE AND FAN DRAFT REGULATOR

would tend to lessen ordinary nuts, the fitting of the crank-shaft brackets to the ram boxes with body-bound bolts, and an increase in the size of the gears in the gear boxes and the shafts and bearings. To insure more certain connection between the cast-iron ram boxes and the tuyere boxes they are now carefully doweled together.

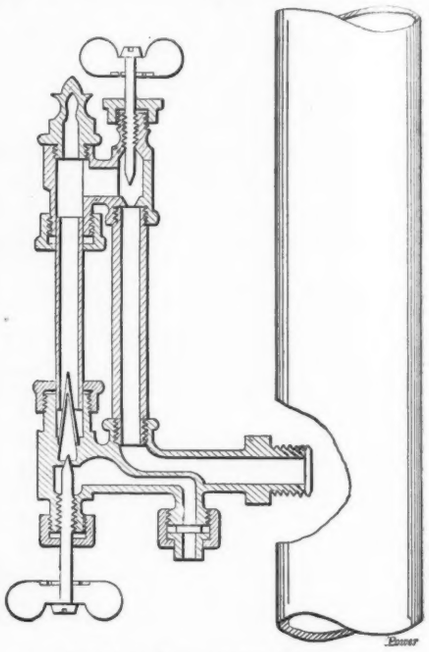
The air for combustion is supplied by a volume blower driven by the same engine which operates the stokers. A change in the rate of feeding coal is therefore accompanied by a change in the rate of supplying air. After the stoker is installed, tests are made to determine the ratio of air and coal to give perfect combustion. The relative speed of the fan and stoker is then fixed; thereafter no further change is necessary. The fireman does not need to worry as to the air supply. In this way a high efficiency is obtained throughout a wide range of load, and errors of judgment are eliminated.

The lever arm of the regulator is connected to the lever arm of a butterfly or chronometer valve in the steam line of the fan engine by means of cables and pulleys. Whenever the steam-pressure fluctuations cause the damper-regulator lever arm to move up or down the fan engine speeds up or slows down accordingly.

The Ashton Sight Feed Lubricating System

This system consists of a central receptacle, or reservoir, from which the cylinder oil is distributed to each cylinder and fed through a sight-feed device shown in section in the illustration. The feeder is screwed into the steam pipe by the nipple. The oil connection from the reservoir is made at the union *E; C* is a valve to be closed should the glass break, and *D* is the needle valve for regulating the flow of oil.

Any number of cylinders may be lubricated by this system, the tank supplied being of a size corresponding with the



ASHTON LUBRICATING SYSTEM

amount of oil to be fed. This system is manufactured by Walter L. Flower & Co., 1023 Chemical building, St. Louis, Mo.

Opening American Museum of Safety

The formal exercises in connection with the permanent exhibition of safety devices of the American Museum of Safety were held on Monday evening, November 21, at the auditorium of the Engineering Societies' building, Philip T. Dodge, presiding. The Museum of Safety is the fourteenth similar organization of which there are twelve in Europe and one in Canada. The object of the museum is the conservation of human life, by means of a permanent exhibit of the best and most practicable safety devices for making safe the dangerous parts of machines and processes.

A number of gold medals were presented and a greeting from President Taft was read to the gathering.

Dr. W. H. Tolman, director of the museum, who had spent the summer in Germany, studying its system of accident prevention, showed by means of lantern slides how the fatherland is saving 50 per cent. of the accidents to labor and told how in the last 25 years Germany had spent 2½ billion dollars in preventing accidents.

T. C. Martin, chairman of the executive committee, made an earnest plea for additional funds to enlarge the exhibition space as the present exhibit hall is already outgrown. Other speakers of prominence responded to the call of Mr. Dodge with a few brief remarks on the work of the museum.

Springfield Convention of Ohio Engineers

While the attendance at the twenty-second meeting of the Ohio Society of Mechanical, Electrical and Steam Engineers, which was held at Springfield, O., November 18 and 19, was not as flattering as at some of the previous gatherings, the amount of interest shown by those present and the entertainment provided by the citizens of Springfield were fully up to the standard of former years. A number of new members were taken in during the session, indicating a healthy growth of the organization.

M. C. Huyette, of Buffalo, presented an illustrated paper on "Automatic Smokeless Furnaces and Smoke Repression," in which a number of cases were cited showing the increased economy which resulted when proper means had been provided to eliminate smoke. Furnaces were classified as:

First. Overfeed, including all hand-fired furnaces and the "sprinkler" and "flipper" mechanical coal-feeding devices on the market.

Second. Front-feed, including all mechanical furnaces with grates inclined from the front toward the bridgewall, and all chain-grate and traveling-grate furnaces.

Third. Under-feed furnaces, where coal is forced upward, the coking of the coal and combustion of fixed carbon and volatile combustible taking place above the green coal.

Fourth. Side-feed, in which the coal is automatically fed at the sides of the furnace and provision is made for mechanically breaking up the coke and removing the ash and refuse to the ashpit.

It was pointed out that the cost of power is the first and important factor in the cost of any manufactured product, and is seldom known with any degree of accuracy. As an illustration of how power production may be brought down to a question of dollars and cents, the performance at a plant in the Union stock yards, Chicago, was given, where a battery of boilers, rated at 400 horsepower each, were worked at 11½ to 40 per cent. above rating, using Illinois screenings costing $1.10 per ton. The boilers gave an average evaporation from and at 212 degrees of 7.58 pounds of water per pound of coal, the cost of steam being $0.07 per 1000 pounds.

The financial damage caused by smoke was also taken up in detail, showing from the reports of smoke inspectors and others interested throughout the Middle West to what enormous figures this damage will run. In the city of Chicago it is estimated that this amounts to $50,000,000 a year or $20 *per capita*.

Another paper, entitled "The Smoke Situation in General," by George R. Bott, was presented on this important subject. The author spoke of the work now being done to prevent smoke and referred to the difficulties encountered in the small one-man plant. A remarkable case was then cited as to what can be done to improve the furnace conditions of a steam-boiler plant. The plant, consisting of three boilers, was located in New York State, and being unable to carry the load, those in charge decided to improve the conditions. A new 150-horsepower boiler and 20-foot additions to the stacks of the boilers were first thought of; however, the improvements were started by equipping the old boilers with shaker grates and an improved smokeless furnace using superheated steam. After some weeks of operation, the changes noted as having taken place were as follows: Coal at $1.50 per ton was used in place of coal costing $2.10 per ton; two men, firing the boilers, working twelve-hour shifts, had taken the place of six men, working two at a time on eight-hour shifts; the average steam pressure had risen from 50 to 80 pounds; steam blowers in the stacks were discontinued and no extensions were necessary; there was also a marked decrease in the amount of coal used.

"Vapor Heating Systems" was the subject of a paper by T. G. Monat, of Cleveland. Such a system was defined as a steam-heating system operating at a pressure of two to three ounces per square inch, with either a single- or double-pipe installation. No vacuum is employed, the air being expelled to the atmosphere by the pressure alone. The system is essentially one for residences or smaller-sized buildings, the advantages claimed for it being lower cost than a hot-water system, no danger from overpressure because of direct connection to atmosphere, no power required to remove the air, no noise in the system, greater economy because of the low temperature of the returns which are consequently in a condition to readily absorb a large amount of heat from the furnace.

W. E. Haswell, chief examiner of engineers for the State of Ohio, addressed the society on the necessity for a State boiler-inspection law and asked for its coöperation along this line. A resolution was passed to appoint a committee of five to confer with the State examiner's office, boiler inspectors and others interested and to lend the aid of the society toward the drafting and passing of a satisfactory law.

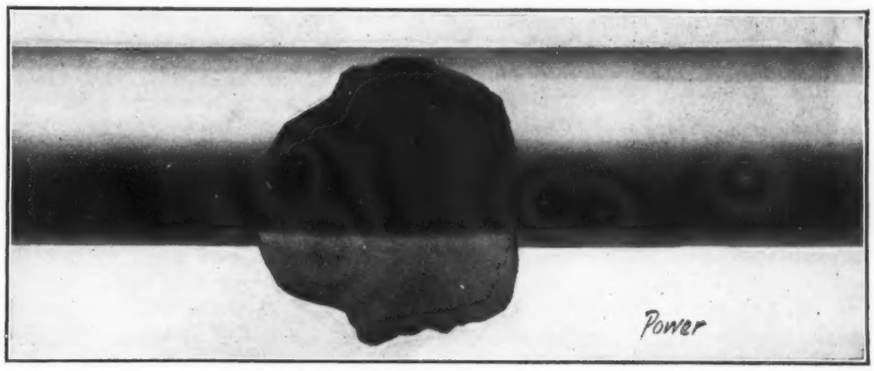
Oscar F. Rabbe, of Toledo, O., was re-elected president; H. L. Patterson, of Youngstown, was elected vice-president and J. J. Hoppes, of Springfield, and Dan Delaney, of Cincinnati, were elected new members of the board of managers. Charles P. Crowe, Ohio State University, Columbus, was retained as acting secretary.

The business sessions were held in the rooms of the Springfield Commercial Club, where everything necessary was provided for the comfort of the visitors. The entertainment program included a number of interesting automobile trips to the various industries of Springfield, the plants visited being the American Seeding Machine Company, Foos Gas Engine Company, Kelley-Springfield Road Roller Company, Kelley Motor Truck Company, Foos Manufacturing Company, Lagonda Manufacturing Company and Hoppes Manufacturing Company. On Saturday the members and visitors were given a luncheon at the Lagonda Club, as the guests of J. J. Hoppes, of the Hoppes Manufacturing Company.

Youngstown was chosen as the place of next meeting, the date of which will be May 18 and 19, 1911.

Boiler Tube Failure

At about 9.15 Monday evening, November 6, a tube in one of the boilers at the Gold street station of the Brooklyn Edison Electric Illuminating Company exploded, painfully, though neither seriously nor dangerously, scalding two men. The tube which failed was the second from the wall in the second row. This boiler had been "let down" during the afternoon for the purpose of putting a new gasket on one of the manhole covers in one of the drums, and at 5 o'clock a fresh fire was started. When the steam pressure had reached 160 pounds, or 20 pounds below the regular working pressure, the failure occurred.



RUPTURED SECTION OF TUBE

The opening in the tube is of a half-oval shape, 4 inches long and 2 wide.

A part of the heated water flashed into steam as it poured out of the hole in the tube. Striking the grates the steam was deflected and driven directly across the 16-foot alleyway between the two rows of boilers, through the open door of the furnace of the boiler opposite. There were several men in this furnace, repairing the brickwork, and it was in attempting to escape by the door through which the steam and water were entering that they were scalded.

The tube was made of charcoal iron No. 10 gage, though at the point of initial rupture it was barely 1/32 inch thick. The boiler has been in use a few months over four years.

Annual Meeting of American Society of Mechanical Engineers

The thirty-second annual meeting of the American Society of Mechanical Engineers will be held in the Engineering Societies building, 29 West Thirty-ninth street, New York, December 6 to 9. Tuesday evening will be the occasion of the annual presidential address and of a reception by the president and the president-elect. The professional papers to be presented are unusual in variety and merit. On Wednesday morning, following the transaction of business, an account of the joint meeting in England will be given by the secretary, Calvin W. Rice; and there will also be a paper by George A. Orrok, of the New York Edison Company, on the transmission of heat in surface condensation. On Wednesday afternoon a session on steam engineering will be held, with a paper on carbon dioxide as an index to combustion, by E. A. Uehling, of the Uehling Instrument Company; two accounts of tests on steam turbines in the locality of San Francisco, one by S. L. Naphtaly, of the City Electric Company, and the other by F. H. Varney, of the Pacific Gas and Electric Company. Other related topics will also be presented. The reception by the local membership will be held on Wednesday evening instead of Thursday evening as heretofore.

On Thursday there will be three sessions: one in the morning upon miscellaneous topics; and in the afternoon two simultaneous sessions, a machine-shop meeting devoted to the subject of grinding with papers by C. H. Norton, of the Norton Grinding Company, W. A. Viall, of the Brown & Sharpe Manufacturing Company, and B. M. W. Hanson, of the Pratt & Whitney Company; and the meeting of the gas-power section, beginning with a paper by E. P. Coleman upon the blast-furnace gas-power installation of the Lackawanna Steel Company.

On Thursday evening there will be an address by Dr. Georg Kerschensteiner, superintendent of schools in Munich, on "Industrial Continuation Schools of Munich." Doctor Kerschensteiner has been foremost among educators in Europe in bringing industrial establishments into coöperative relations with the public-school system. So much importance is attached to this address that it is proposed to make the session a joint meeting with the National Society for the Promotion of Industrial Education, the American Institute of Mining Engineers and the American Institute of Electrical Engineers.

Friday will be devoted to excursions to points of engineering interest, thus making it possible for the out-of-town members to become more familiar with New York, as well as concluding the meeting in an informal way which should give opportunity for increased acquaintanceship in the society.

Correction

In the article on "Exhaust Steam Regenerators," page 1893, November 1 number, the author overlooked a clerical error in the statement that the first Rateau installation was made in 1890. The first installation was put in commercial use in 1902.

PERSONAL

C. W. Clarke has resigned his present position as steam engineer with the New York Central & Hudson River Railroad Company in order to accept an appointment with the Stone & Webster Engineering Corporation, of Boston, the change to take effect December 1, 1910.

NEW PUBLICATION

"The Effect of Keyways on the Strength of Shafts," by Herbert F. Moore, has just been issued as Bulletin No. 42 of the engineering-experiment station of the University of Illinois. This bulletin records the results of tests made to determine the relative strength of solid shafts and shafts with keyways. Various sizes of shafts were tested, and for each size of shaft the weakening effect of keyways of several proportions was determined. Tests were made on shafts subjected to twisting only and on shafts subjected to twisting and bending at the same time. The results show that the weakening effect of keyways of the usual proportions upon the strength of shafts is considerable, a square keyway of the usual size causing a reduction of about one-sixth in the strength of the shaft. Formulas and a diagram are given for determining the effect of keyways of various proportions. The effect of keyways for the Woodruff system of keying is discussed. The results of the tests are summarized in a table showing the power which can be transmitted by various sizes of shafts with keyways of usual proportions.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Frank W. Frueauff; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held September, 1911, in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

INTERNAL COMBUSTION ENGINEERS' ASSOCIATION.

Pres., Arthur J. Frith; sec., Charles Kratsch, 416 W. Indiana St., Chicago. Meetings the second Friday in each month at Fraternity Halls, Chicago.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; acting sec., Charles P. Crowe, Ohio State University, Columbus, Ohio. Next meeting, Youngstown, Ohio, May 18 and 19, 1911.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.